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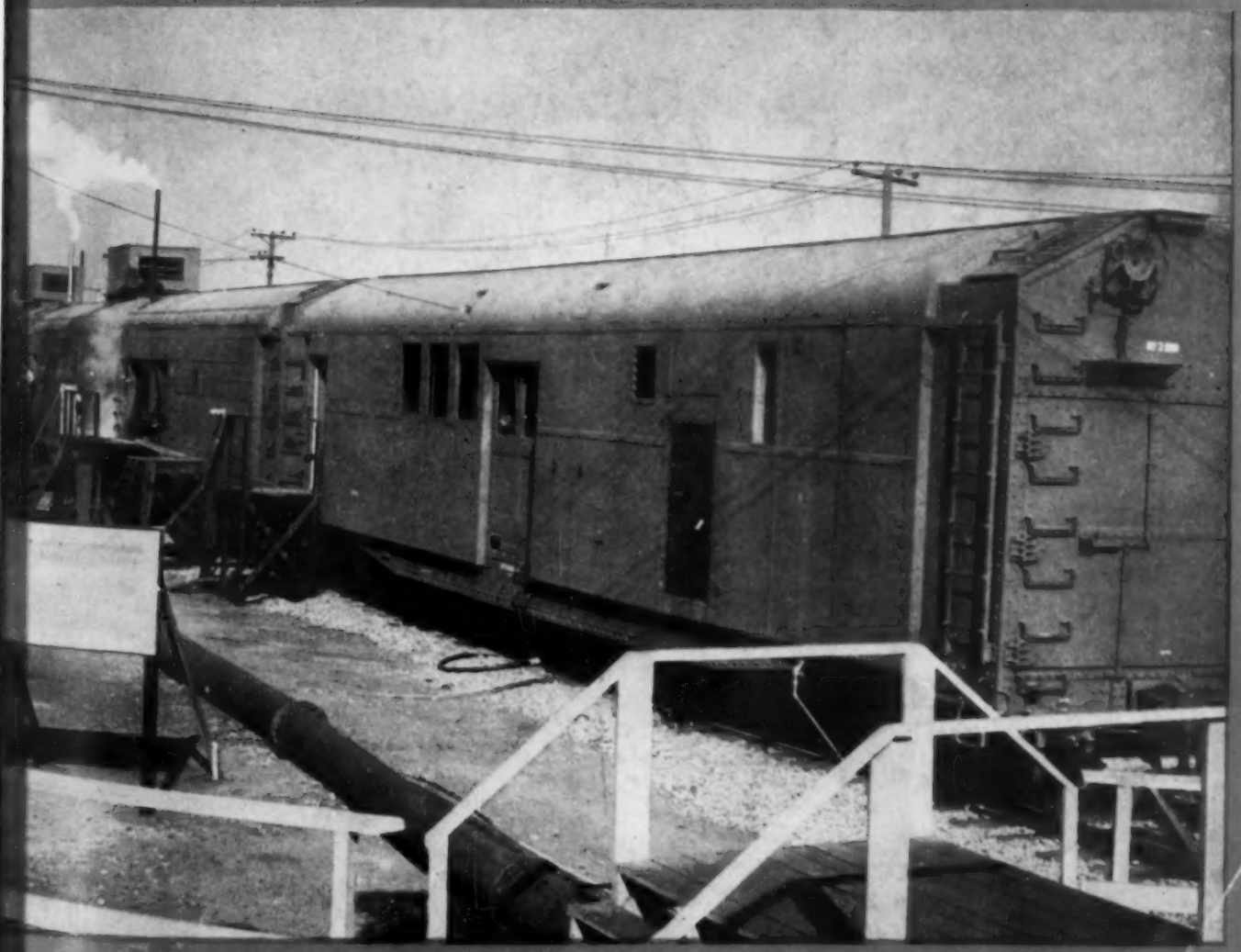
Heating Piping & Air Conditioning

DESIGN
INSTALLATION
OPERATION
MAINTENANCE

SEPTEMBER ★ 1944

INDUSTRIAL
COMMERCIAL
AND LARGE
BUILDINGS

537: *Journal Section of the American Society of Heating and Ventilating Engineers*



Mobile steam power plant on test. V. A. Elliott describes the piping connections for such units in his article this month

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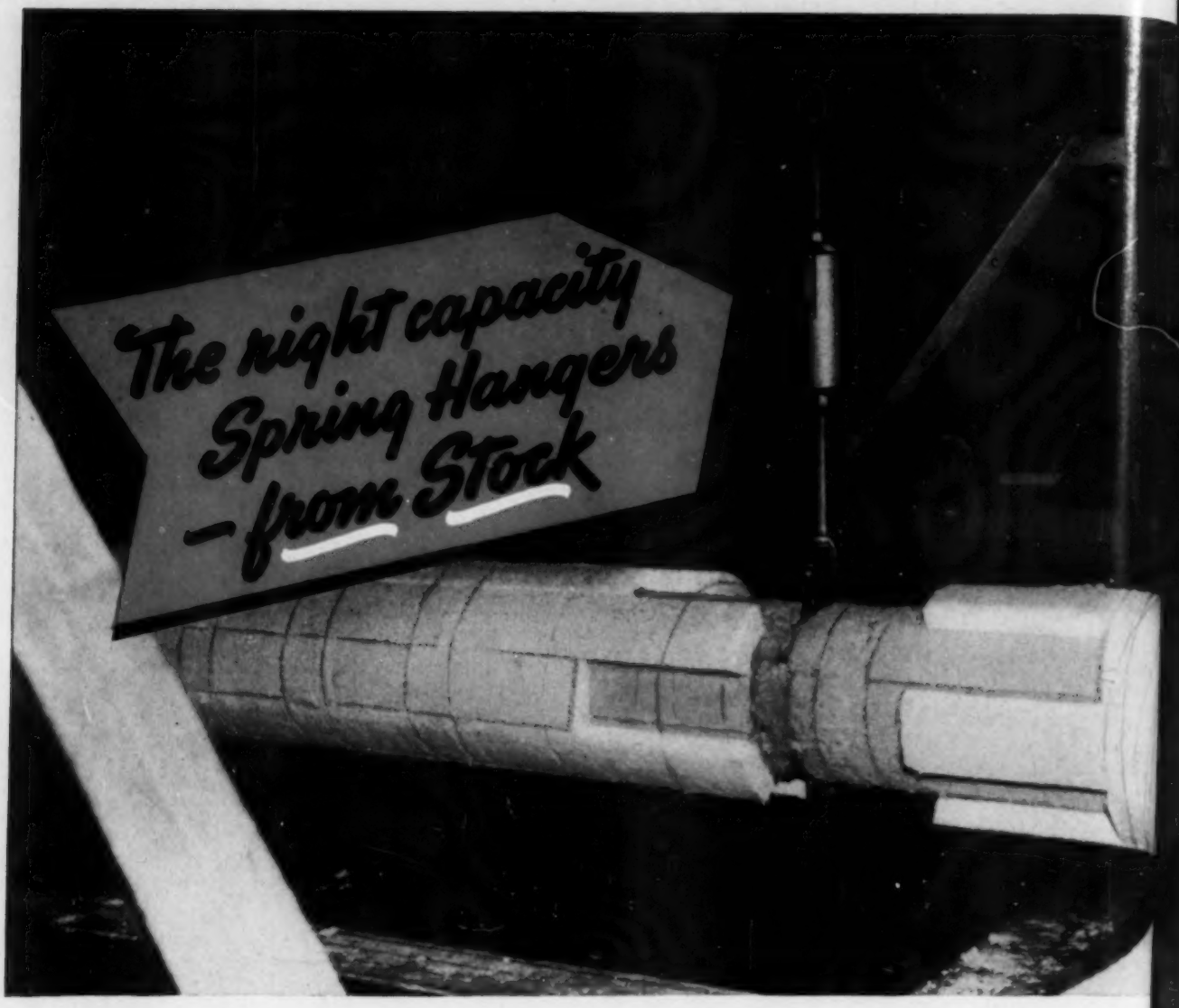

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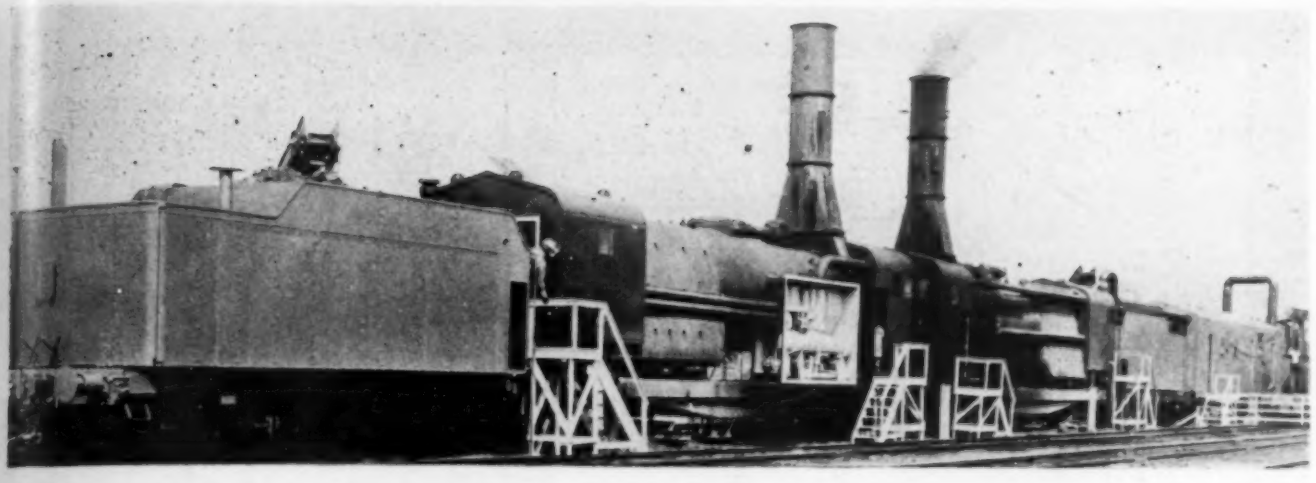

How Piping Connections Are Made on Mobile Steam Power Plants

DESIGN of piping systems for mobile steam power plants offers a number of problems that are not encountered in stationary practice. It is desirable to keep all of the piping within the railroad clearances so that it will not be necessary to remove sections of piping during transportation. Each of the intercar connections offers many problems as provision must be made to allow for misalignment of the cars, unequal wheel wear, variable loading of cars, and connections for operation on curved tracks. Usually it is necessary to have runs of piping through a number of cars, giving several inches of thermal expansion. It is desirable to provide flexibility in the intercar connections, leaving straight runs of piping in the car, which allows more space around the equipment.

The piping systems discussed here have been used on mobile steam power plants of 10,000 kw, 3000 kw, and 1000 kw capacity. Two 10,000 kw plants are now in operation at the customer's site. Two of the 3000 kw plants have been tested on a railroad siding with a load run of 72 hr duration; during the assembly of the train as well as in the operation, the piping systems met every expectation. No drawings of the piping are shown for the 1000 kw units, as the piping connections are similar to those used on the 3000 and 10,000 kw units.

Corrugated Piping Reduces Expansion Loop Dimensions

Fig. 1 shows the main steam piping for the 3000 kw mobile steam power plants. The design conditions for the steam piping were 300 psi gage and 750 F. The problem consists of connecting the two boiler cars and then routing the 8 in. line over the tender through the switchgear car into the turbine-generator car. The piping is anchored in four positions and a major portion of the flexibility is provided in the intercar connections. Corrugated piping is used to reduce the physical dimensions of the expansion loops.

The corrugated connections between boiler cars is provided with van stone flanges in a horizontal plane to allow for lateral misalignment of the cars and to take care of setups on curved tracks. Vertical adjustment of the pipes is provided by adjustable guides which support the piping in the stack end of the boiler cars. The flanges can be properly made up by varying the distance between cars.

The connection between boiler No. 2 and the tender No. 2 is provided with van stone flanges. This arrangement gives a pivot point for setups on tracks that are not straight.

The piping between the tender and the switchgear car utilizes corrugated pipe, van stone

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V. A. Elliott Describes Piping Details for 10,000, 3000, and 1000 Kw Railroad Units

Mobile steam power plant units, completely mounted on railroad cars, are used for many wartime services, including the rehabilitation of areas laid waste by the enemy. The design of the piping for these plants offers a number of problems not encountered in stationary practice. It is desirable to keep the piping within railroad clearances so that sections of the piping need not be removed during transportation. Provision must be made in the intercar connections to allow for misalignment of the cars, unequal wheel wear, variable car loading, operation on curved tracks, etc. How this is done is described here by Mr. Elliott, of the construction engineering division of the General Electric Company.

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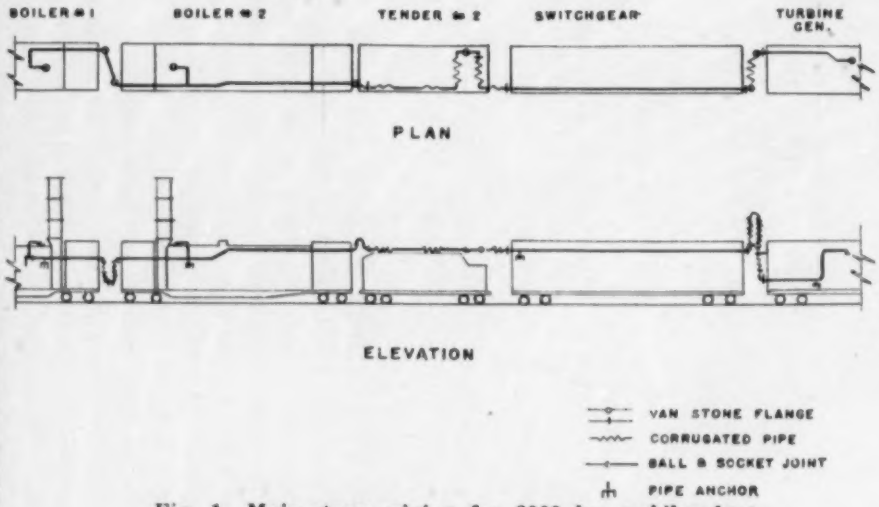

Fig. 1—Main steam piping for 3000 kw mobile plant

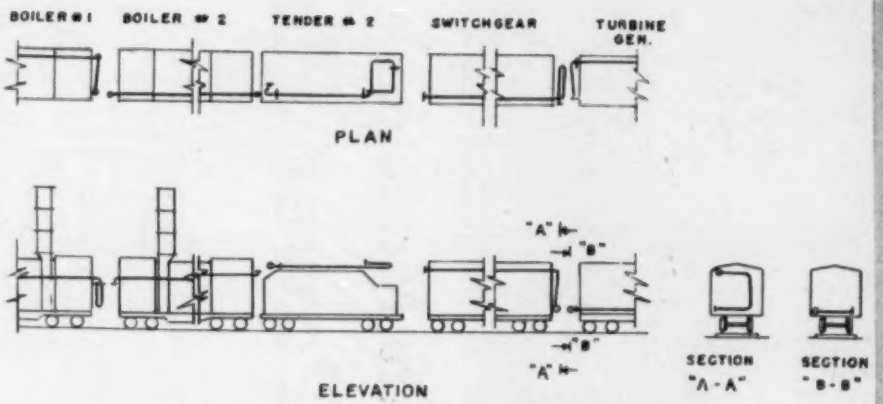

Fig. 2—Method of transporting intercar connections

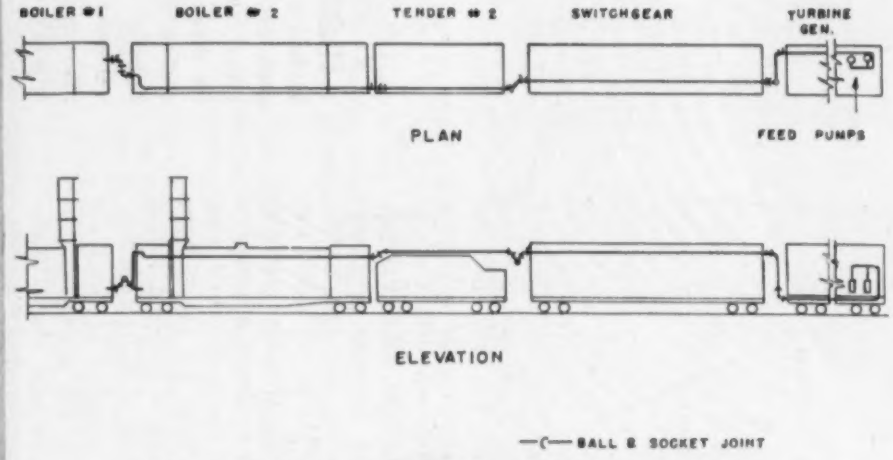

Fig. 3—Boiler feedwater piping, 3000 kw mobile plant

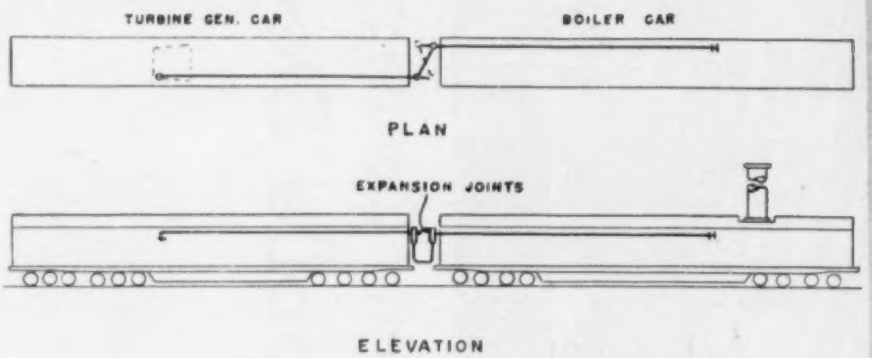

Fig. 4—Main steam piping for the 10,000 kw plant. Dotted lines between cars show hot position of piping

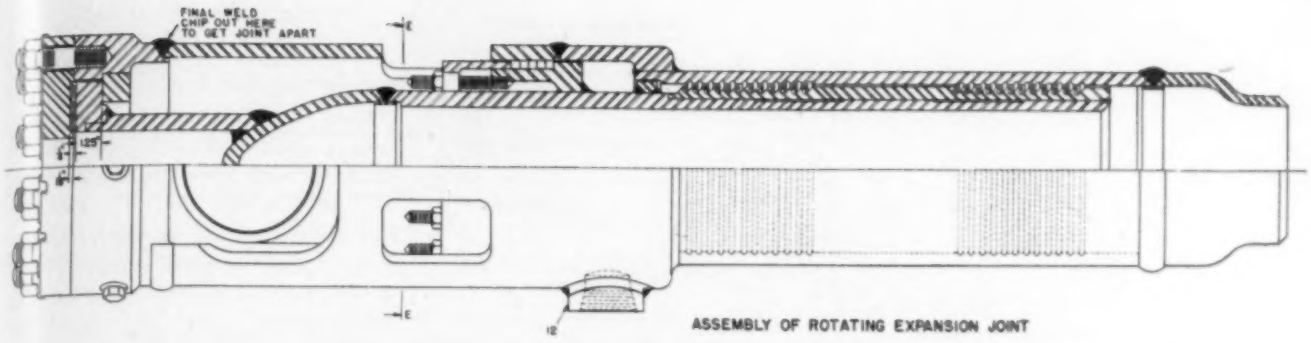

Fig. 5—Rotating type expansion joint for 10,000 kw unit

flanges, and a ball and socket joint. The combination of the above features gives flexibility in all directions to allow for vertical and lateral car misalignment. The ball and socket joint also increases considerably the flexibility of the steam loop as far as expansion is concerned.

The connection between the switchgear and the turbine-generator cars is made up of corrugated piping with two van stone flanges in both the vertical and horizontal planes. The four van stone flanges give ample adjustment for making up the flanges when the train is being set up for

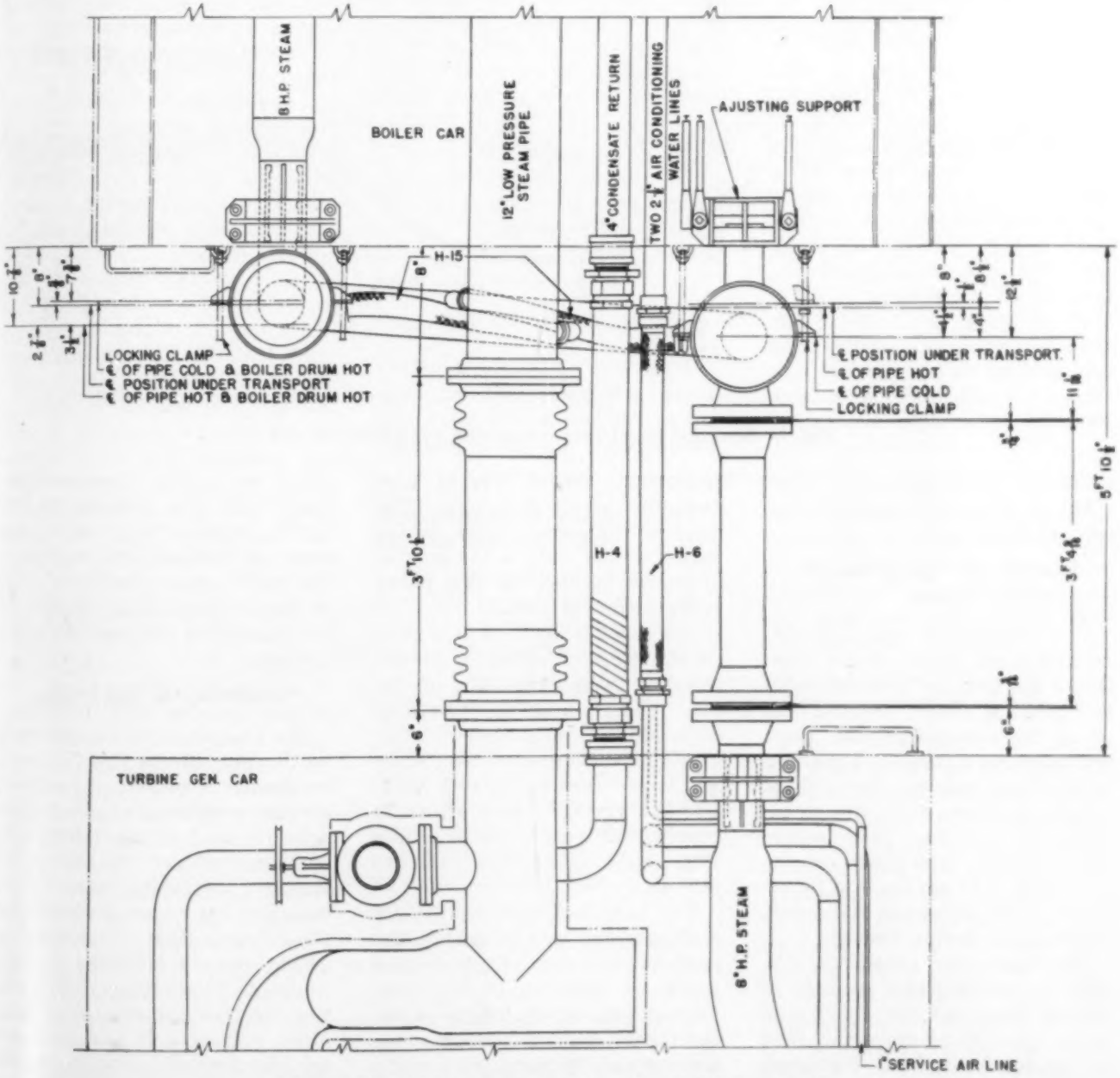

Fig. 6—Plan of intercar connections for 10,000 kw plant

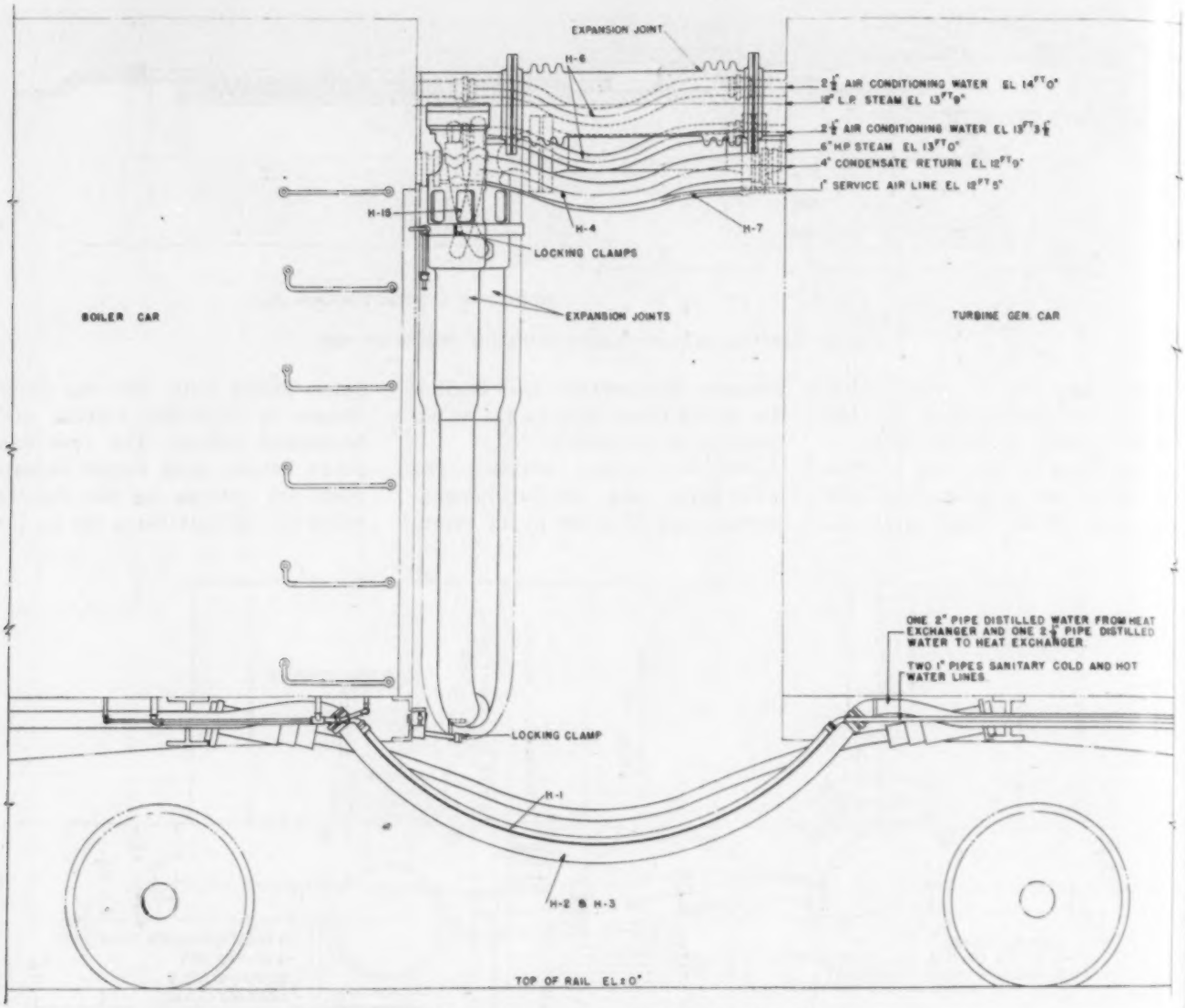
EXPANSION JOINT
H-6
2½" AIR CONDITIONING WATER EL 14FT 0"
12" L.P. STEAM EL 13FT 9"
2½" AIR CONDITIONING WATER EL 13FT 3½"
6" HP STEAM EL 13FT 0"
4" CONDENSATE RETURN EL 12FT 9"
1" SERVICE AIR LINE EL 12FT 5"
H-18
H-4
H-7
LOCKING CLAMPS
BOILER CAR
EXPANSION JOINTS
TURBINE GEN. CAR
ONE 2" PIPE DISTILLED WATER FROM HEAT EXCHANGER AND ONE 2½" PIPE DISTILLED WATER TO HEAT EXCHANGER
TWO 1" PIPES SANITARY COLD AND HOT WATER LINES
LOCKING CLAMP
H-1
H-2 & H-3
TOP OF RAIL EL±0"

Fig. 7—Elevation of intercar connections for 10,000 kw unit

operation. Two springs are provided to stay the expansion loop against wind loads.

Position of Piping During Transit

The position of the interconnecting steam piping during transit for the 3000 kw units is shown in Fig. 2. It should be noted that all of the connections fold about the van stone flanges to a position within the clearance line without removing them from their respective positions. This feature offers definite advantages as the connections do not have to be removed from the piping and stored inside a car during transit.

The feedwater piping for the 3000 kw mobile plants operates at 400 psi gage and 220 F. Fig. 3 shows the piping from the feed pumps to the boilers. Universal joints are used for each of the connections between cars to give flexibility in all directions. The intercar connections are transported on the ends of the cars in a similar manner to the steam piping shown in Fig. 2.

The small water and drip lines on the 3000 kw mobile units are installed in the side sills of the cars. The pipes are insulated and enclosed by a metal cover. The intercar connections are hoses which are removed during transit. The heating system utilizes a conventional train heating line with hose connections at the couplers.

The 3000 kw plants have three cooling tower cars to provide the condensing water. Each cooling tower car has two 14 in. water headers running the length of the car. These headers are connected between cars by means of a multi-arch rubber expansion joint which may be left connected during transit. Fig. 8 shows the connection between the 14 in. cooling tower headers and the condenser. The multi-arch rubber expansion joints are identical to those used for connecting the cooling tower headers.

Insulation of the Piping

The insulation of the piping in the cars offered no unusual problems. However, it required special consideration to provide access to each of the flanged connections between the cars. On the 3000 kw plants, there are removable insulating blankets for the flanges and connections to which access is necessary for disconnecting the piping in preparation for transportation. The small hose connections between cars are also provided with removable insulating blankets.

The 10,000 Kw Units

Fig. 4 shows a schematic diagram of the main steam piping used on two 10,000 kw mobile plants, the design steam conditions for which were 560 psi gage and 825 F. The steam line has straight runs in the cars with the flexibility provided in the connection between the cars. The line is anchored at the superheater header and the turbine throttle. The thermal expansion is taken care of by two rotating type expansion joints in the intercar connection. As the steam line expands, the intercar connection takes a position as indicated by the dotted lines of Fig. 4.

Fig. 5 is a cross section of the rotating type expansion joint for the 10,000 kw plants. The expansion joint essentially is made up of two sleeves with one arranged to rotate within the other. In order not to pack the joint against full steam pressure, a labyrinth packing is provided to allow the steam to expand from 560 psi to 2 psi gage through a leakoff connection to the 2 psi steam header. The clearances are such as to keep the quantity of leakoff steam to the minimum. A ring type packing is used to seal the joint from 2 psi to atmosphere. A bearing is provided in the head of the joint to take the pressure thrust between the two rotating elements.

Fig. 6 shows a plan view of the details of the connections between the turbine and boiler cars on the 10,000 kw units. To compensate for car misalignment, an adjustable support is provided on one of the expansion joints to give both vertical and lateral movement. The rotating expansion joints and the connecting loop are carried in transit by tightening the locking clamps. A copper expansion joint is used in the 12 in. low pressure steam header between cars. For connecting the small pipes, flexible metallic hoses are used. The copper expansion joint and the hoses are stored in the parts car during transit. Fig. 7 shows an elevation of the intercar connections on the 10,000 kw plant.

The 10,000 kw mobile plants must obtain circulating water from a river or other source. A circulating pump and portable piping is provided with each power plant. The portable piping consists of approximately 500 ft of 14 in. flanged duck and rubber hose for the pump discharge and 60 ft of 16 in. hose for the pump suction.

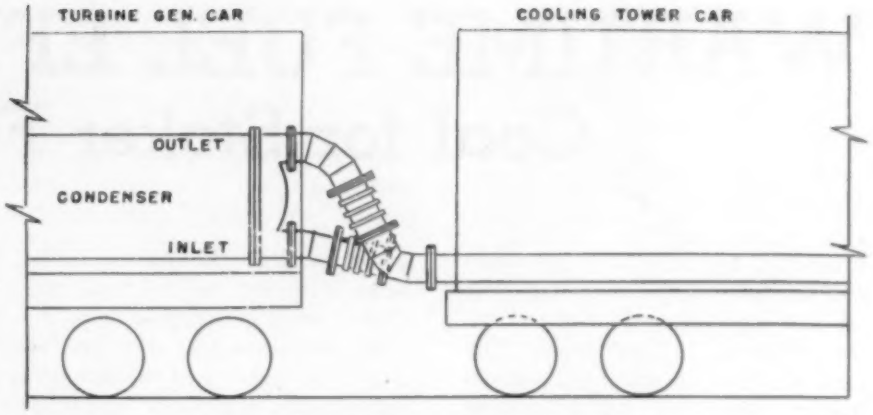

Fig. 8 — Circulating water connection to condenser, 3000 kw mobile steam power unit

The piping between cars on the 10,000 kw mobile plants has the insulation protected with a metal cover. Access is made to the flanges by unbolting removable insulated metal covers.

Heating and Air Conditioning

The temperature of the operating space in the 10,000 kw mobile steam power plants is controlled by circulating the air through heat exchangers mounted on the roof of the car. The roof unit includes a fan, a heating coil, and a cooling coil. Damper control is provided to allow recirculation of the air as well as to permit outside air being taken in from 0 to 100 per cent of fan capacity. For winter operation the air is heated by the heating coil, which obtains steam from the train heating line. For warm weather operation the air is chilled by the cooling coils which are supplied with cold water from a steam jet refrigeration system of 70 tons capacity situated in the turbine car. The cooling water is piped through the boiler and turbine cars with the connections between cars made with hoses. The jets of the refrigerator exhaust into the main turbine condenser.

CONSTRUCTION PRICE REGULATION REVISED

The construction industry may now add to presently established ceilings for construction services, increases in wage costs since October 3, 1942, that have been approved or authorized by the Wage Adjustment Board, National War Labor Board, or Economic Stabilization Director, the Office of Price Administration announced last month. A number of other changes in the regulation covering the construction industry have also been made.

The construction industry has been under price control since early 1942, and there are ceiling prices on all building and installation services. Revised MPR No. 251, effective August 26, the agency emphasized, simply revised the regulation for the purpose of liberalizing its provisions to better fit the needs of contractors. Procedural and record requirements, now have been reduced to an absolute minimum, OPA said.

29 LIMITATION ORDERS AMENDED

Twenty-nine limitation orders governing production of consumers' durable products have been amended to indicate specifically in the orders themselves that the products they cover are subject to the "spot authorization" procedure issued on August 15, 1944, the War Production Board announced August 22. This action merely removes provisions in the orders that might appear to conflict with priorities regulation No. 25 which contains the rules governing the "spot procedure."

WARTIME FUEL EFFICIENCY
Coal for Stoker Firing

As a contribution to the national fuel efficiency program sponsored by the Bureau of Mines, we present a group of articles by Kalman Steiner, of which this is the third and last, discussing some fundamentals of the nature and utilization of coal. The Bureau of Mines is disseminating information on improvements in operating methods, so it was felt that operating engineers would be aided by a review of the basic technology upon which those methods are based. The article in the June issue indicated the significance of coal analyses, Btu values, and more general aspects of coal composition, as well as the basic concepts of coal combustion. The August installment was devoted to the technique of hand firing. The properties of coal which affect its use in mechanical stokers are considered in this month's article

IT WILL BE helpful, in considering not only stoker fuels but coal for all other uses, to understand the meaning of coal *rank*. The rank of a coal designates its classification according to a system based on the proximate analysis. All coals fall into four major *classes*, which again are subdivided into *groups*. The four classes are known as *anthracite*, *bituminous*, *subbituminous*, and *lignitic*. Before discussion of the characteristics of these coals can be attempted, it will be necessary to define a few technical terms. These definitions are a continuation of those given in the June article.

Laboratory Analyses of Coal

There are at least five bases for expressing the results of a laboratory analysis of coal: (1) as received, (2) moist—really meaning air dried, (3) dry—all water expelled by heat, (4) moisture and ash free, and (5) moisture and mineral matter free. In (1), each constituent of the analysis is reported as a percentage of the coal as it existed when received at the laboratory. The moist basis (2) is supposed to represent what is in the coal after it is freed of surface moisture by allowing it to air dry until no water is apparent to visual inspection. The dry basis (3) reports each constituent as the percentage of what remains after heating the sample at 221 F for 1 hr. This procedure is supposed to eliminate even the water inherent to the coal substance as it is removed from the mine. Moisture and ash free (4) shows all water and the ash subtracted, leaving only combustibles. Finally, the basis (5) is an attempt to correct for changes occurring in the mineral constituents during combustion and accounts for sulphur in both the ash and organic substance.

In the laboratory determination of volatile matter (see June issue) there is formed in the crucible a coke button residue. A tentative ASTM specification (D 388-36T) states "Coals which in the volatile matter determination produce either an agglomerate button that will support a 500 gram weight without pulverizing, or a button showing a swelling or cell structure, shall be considered agglomerating from the viewpoint of classification." Thus the *agglomerating* properties of a coal are related to its caking or coke forming properties, and are indicative of its performance during combustion. Some writers use the term *agglutinating* instead of agglomerating.

The tendency of a coal to disintegrate, or form smaller particles —even dust—is called *weathering* or *slacking*, and is quite undesirable. In coal utilization, it is a distinct advantage that the coal resist disintegration or breaking up. It is obvious that if coal has been screened at the mine into certain sizes, and has a marked tendency toward weathering or slacking, upon arrival at its destination it will no longer consist only of the sizes established by screening.

The coal class anthracite comprises three groups, known as *meta-anthracite*, *anthracite*, and *semi-anthracite*, all of which are ranked on the dry and mineral matter free basis as follows: meta-anthracite—fixed carbon (FC) 98 per cent or over and volatile matter (VM) 2 per cent or less; anthracite—FC 92 to 98 per cent and VM 2 to 8 per cent; semi-anthracite—FC 86 to 92 per cent and VM 8 to 14 per cent. Coals to be placed in class anthracite must be nonagglomerating. Should a coal otherwise have the properties of one of these three groups but be agglomerating, it is classified as low volatile bituminous.

Coal Classes Further Divided Into Groups

Coals in class bituminous are subdivided into five groups, and subbituminous into three, as shown in the accompanying table. Class bituminous coals must be either agglomerating or nonweathering, while to be classified subbituminous coals must be both

weathering and nonagglomerating. It will be noted that the low volatile and medium volatile groups are classified similarly to the anthracite class, i.e., on the basis of dry and mineral matter free fixed carbon and volatile matter. The group high volatile A bituminous is ranked in part similarly, but must also qualify on 14,000 Btu per lb minimum, moist basis, except that if the fixed carbon is 69 per cent or more the Btu requirement is disregarded. The groups high volatile B and C, and subbituminous A, B, and C are ranked entirely on the basis of moist Btu content.

In class lignitic there are two groups, referred to as lignite and brown coal, respectively; both have less than 8300 moist Btu value. In physical form, lignite is required to be consolidated, while brown coal is unconsolidated.

In referring to coal analyses on the dry and mineral matter free basis, it must be borne in mind that all the values shown are considerably higher than actually prevail in the "as received" coal. A coal that contains 2 per cent surface moisture, 3 per cent inherent moisture, 7 per cent ash, and 2 per cent sulfur, and has a calorific value of 12,500 Btu per lb, all as received, on the basis moist (with mineral matter) would show 12,500/0.98 or 12,755 Btu, and on the basis moist and mineral matter free would show 13,849 Btu per lb, from the following American Society for Testing Materials formula:

$$\text{Moist, Mm-free Btu} = \frac{\text{Btu} \times 100}{100 - (1.1 \times \%\text{ash} + 0.1 \times \%\text{sulfur})}$$

Since moisture and ash content of a coal can be and are affected by mining and handling methods, as was pointed out in the June issue, the method of reducing data to the dry and mineral matter free basis compares the properties of the coal substance itself, and is therefore the logical basis for classification of coals.

Stokers Not Bothered by Volatile Content

The preferred coals for hand firing are the low volatiles, including all three groups of class anthracite and the group low volatile bituminous. As the volatile matter increases above 22 per cent, the difficulty of hand firing increases, insofar as smokeless combustion is concerned. Mechanical stokers are not particularly bothered by volatile content. The standard forms of stokers can burn coal of almost any volatility or rank, providing of course that the furnace setting or combustion chamber is designed properly and in accordance to the load to be carried. The factors determining what coal to use in a particular installation or stoker are not reflected in the rank classification, except as to agglomerating characteristics.

Since coal must be purchased on an "as received" basis, it is important to know how much heat will be obtained per dollar expended; essentially, coal is a form of energy and its net value is best expressed in terms of Btu per dollar. Yet it cannot be said the coal giving the most Btu's per dollar is the most economical in any given case, because handling and operating costs are a factor, and some coals by virtue of their low fusing ash or other features may involve high handling charges. The only definite way to establish the economy of a coal is in terms of cost of 1000 lb of steam, including all fuel, labor, and overhead.

In a general way, coals vary from highest rank toward lowest from the anthracite fields of Pennsylvania to the lignites of North Dakota. There are, however, some fair coals, mostly subbituminous, in the state of Washington. Ash and sulfur tend to increase in passing from West Virginia low volatile bituminous to Iowa subbituminous, and calorific value to decrease. But a high rank eastern bituminous may have a higher percentage of ash than a midwestern low rank coal; however, because of the higher heat content of the pure

COAL RANK CLASSIFICATION

BITUMINOUS - CLASS II, AND SUBBITUMINOUS - CLASS III

Class	Group	Limits of fixed carbon or Btu, Mineral matter free basis
II	1. Low volatile bituminous	Dry FC, 78 to 86%; dry VM, 14 to 22%
	2. Medium volatile bituminous	Dry FC, 69 to 78%; dry VM, 22 to 31%
	3. High volatile A bituminous	Dry FC, under 69%; dry VM, over 31%; moist Btu, 14,000 min.
	4. High volatile B bituminous	Moist Btu, 13,000 to 14,000
	5. High volatile C bituminous	Moist Btu, 11,000 to 13,000
III	1. Subbituminous A	Moist Btu, 11,000 to 13,000
	2. Subbituminous B	Moist Btu, 9,500 to 11,000
	3. Subbituminous C	Moist Btu, 8,300 to 9,500

FC = fixed carbon. VM = volatile matter.

coal substance it may exceed the midwestern coal in Btu per lb as received.

Properties of the Ash

Some points relative to *ash characteristics* were brought out in the June article. Probably, next to calorific value, the properties of the ash are the most important consideration in evaluation of a coal for stoker firing (if, indeed, not the most important) from the viewpoint that a high calorific content is of little weight if the ash fuses and fouls up the tuyeres. Just what behavior is required of an ash depends upon the kind of stoker involved, and the percentage of rating being developed. For commercial stokers of moderate size, using front-door removal of clinker, the ash must clinker in order that it may be removed from the firebox. But if the ash fuses too readily it will run to the bottom of the retort and into the tuyeres. If the ash softening temperature is too high, the ash will not clinker properly and will offer much difficulty in its removal from the firebox. It has been reported that larger stokers equipped with water cooled tuyeres have successfully burned coal at rates up to 52 lb per hr for which the ash softening temperature was 1900 F. In other types of stokers—chain grate, overfeed, spreader—the ash softening temperature is not so critical.

Coking and Caking Properties of a Coal

The *coking* or *caking* properties of a coal are important on underfeed and overfeed stokers, especially important on chain grates, and of no consequence on spreader stokers. High rank eastern coals are especially well adapted to underfeed and overfeed stokers, the former being the predominating type used. In thick fuel beds operating with high air pressure, underfeeds will attain sustained combustion rates of 60 lb per sq ft and even 70 at peak load for short intervals, the fuel beds being 24 in. and more in thickness. Coking coals require agitation of the fuel bed, such as prevails in underfeed and overfeed stokers. Chain grates, unless specially equipped, do not agitate the fuel bed, and so caking coals block the passage of combustion air through the fuel bed except where crevices develop, localizing contact of air and fuel, resulting in high excesses of air rushing

through the cracks and unburned fuel being rejected to the ashpit. The noncoking coals of the middle west, mostly high volatile bituminous and subbituminous, are good fuels for chain grate stokers. Spreader stokers burn a considerable portion of the fuel, especially the fines, in suspension, and so the caking quality is unimportant.

Ignition Temperature

The *ignition temperature* of a coal is the lowest temperature at which it will begin to burn. As a coal is heated, it first loses moisture and then volatile combustibles. Evolution of the combustibles begins at about 650 F. The ignition temperature of the coke formed by the fixed carbon of bituminous coal is about 765 F, while the minimum temperature at which the combustible gases will ignite is over 1100 F. Of course, gases distilled from the coal rise immediately to zones of higher temperature and are there ignited, but the fixed carbon must be pushed or moved onward to the hotter zone before it will ignite.

Many chain grate and overfeed stokers provide an ignition arch of refractory over the point of entrance of the coal into the furnace. By radiating heat into the green coal, these ignition arches promote rapid distillation of the volatiles and subsequent ignition of the coke. Underfeed stokers are not so critical to ignition temperatures, although if operating at light load with coal of high ignition point, the plane of ignition will rise upward in the retort and may upset the equilibrium or average operation. That is to say, as the ignition temperature rises, the thickness of the fuel bed in an underfeed stoker increases. Since most stokers move coal through the firebox at a predetermined rate, for any given load condition, the coal is in the furnace only a limited period of time. The sooner ignition occurs, therefore, the better are the chances the coal will be completely burned during its passage through the furnace.

Friability of Coal and Its Importance

Mention has already been made of the property of weathering, which might be called natural crumbling and is primarily due to evaporation of inherent moisture. An allied coal property is called *friability;* it implies likelihood of the coal cracking or breaking up in processes of handling. The converse is of course size stability, or the property of resisting breakage during shipment and other handling. Friability is akin to brittleness. Coal of low friability or good size stability is likely to arrive at its destination in a state of aggregation resembling the preparation given it at the mine, and to maintain that size consist when subject to the mechanical attrition of conveying and feeding equipment. Resistance toward shattering is especially significant in underfeed stokers. Experience has shown that the larger sizes of coal entering the auger of the screw conveyor are prone to break down to smaller particles. The importance of coal size in underfeed operation will be brought out below.

There are two forms of laboratory test for friability. The drop shatter test measures the reduction in coal sizes after dropping a sample twice through a 6 ft height onto a steel or cast iron

plate. This test gives some index of resistance to breakage in the ordinary operations of preparing and transporting lump coal. The tumbler test is conducted by placing a sample inside a cylindrical vessel that is equipped with longitudinal angle irons that will raise the coal as the cylinder rotates, and drop it again as the angle continues upward, the cylinder being 7¼ in. in both diameter and length. The amount of reduction in size during this test is considered an indication of stability during mechanical handling of the coal in the boiler room equipment.

Because the screw conveyor, if used as the feeding method in an underfeed stoker, can exert crushing strength upon lumps of coal, considerable investigation has been made of the relation between size and performance of this type of stoker. It has been established: (1) the percentage reduction of coal size is definitely related to the hardness or friability index, (2) the larger sizes are broken up before reaching the retort, (3) larger sizes tend to increase segregation of coal in the hopper and in the retort, (4) a considerable proportion of lump coal is actually reduced to fines by the time it reaches the retort, (5) avoidance of the larger sizes in the coal placed in the hopper reduces the work of the mechanism (otherwise required to break the large lumps) and gives more uniform distribution in the hopper and the retort. Furthermore, the smaller the average size of the coal pieces, the more uniform will be the rate of feeding of coal to the retort. All things considered, therefore, there are no good reasons for using larger sizes of coal in an underfeed stoker, and several good reasons for using mixtures of fines and smaller sizes.

Experience the Best Teacher

All coal authorities recognize the difficulty of laying down rules for either the selection or use of a coal to suit any particular plant. Only actual experience can give the final answer as to which is the best of possible choices of fuel, or what are the best conditions under which to burn any one fuel. The foregoing review of technical factors should assist the operating engineer in studying and analyzing the performance results. A few facts have, however, been fairly well established with regard to plant operation. For one thing, it is seldom that much benefit is derived from switching rapidly from one coal to another. A boiler room staff requires some time to become familiar with the characteristics of a given fuel. The fireman learns much from watching his fire, observing the caking tendencies, the fuel bed conditions, slagging and clinkering. Sometimes the moisture content makes a great difference in porosity of the coked fuel, and he therefore might find it expedient to dampen the coal, especially the fines, before firing. Furthermore, a short test of a coal is not fair to the fuel or the plant; it might not be of sufficient duration to cover a real cross section of load conditions, but might catch only a period of low or high load, or rapidly fluctuating load, while the normal plant condition might on the average be entirely different.

Blending of Coals

The blending of coals is not a highly developed art, especially in this country. Blending was generally discouraged. Every coal has its own burning rate, and to blend coals so intimately that a fuel bed would burn uniformly is hardly to be expected. The ill effects of having one portion of a fuel bed burn faster than another are obvious—holes develop in the fuel bed, air distribution is uneven, clinkering not uniform, etc. Yet in spite of all these reasons for not blending, wartime conditions sometimes require it. Transportation congestion and other disturbances due to all out war production may restrict the amount of coal available from usual sources and force a plant to resort to blending. If so, "extender" coal should be selected to have chemical and physical properties as close as possible to the usual supply. In some cases, however, it is possible to use a mixture of high volatile bituminous and anthracite to replace a low volatile bituminous. Whether such practice can be considered, practically, depends of course upon the relative freight rates of the several coals. If adopted, the mixing is best done, or at least started, at the yard. The anthracite can be spread as the bottom layer in a truck and covered with the bituminous. Upon being dumped into the user's bin, fairly good mixing results. If coal is purchased by carload, alternate cars of bituminous and anthracite should be unloaded into the storage piles.

Comparing Two Fuels

In comparing relative results between two different fuels, as mentioned above, it is necessary to consider all items that enter into the cost of producing steam. It is not difficult to reduce to a common basis such expenses as labor for firing, cleaning and maintaining stoker equipment, etc., for one has but to divide the total of such expenses by the total tons of coal burned during the same length of time. To compare the prices of the two fuels, it is necessary to know the Btu value and combustion efficiency obtained with each. With data in the following form, a calculation will show at what price a second fuel must be bought to equal an established steam cost.

If D_a = price of coal a, dollars per ton; H_a = Btu per lb, coal a; E_a = Combustion efficiency secured with coal a; H_b = Btu per lb, coal b; E_b = Combustion efficiency secured with coal b; then

$$D_b = \frac{D_a H_b E_b}{H_a E_a} = \text{dollars per ton at which coal } b \text{ should be bought for equivalent combustion results}$$

The value of D_b thus secured should be corrected for the other direct expense items as calculated above.

There is one factor in plant operation that cannot be reduced to a simple formula, and that is uniformity of steam pressure. In a number of plants, such as laundries and other process loads, nonuniform steam pressure can seriously affect continuity and quality in production.

Design and Control of year 'round comfort Air Conditioning Systems

THE TEMPERATURE control system required for a primary air system is much more complex than for a straight reheating system. One reason for this is, of course, that the control system must swing *automatically* from the summer cycle to the intermediate cycle, or from the intermediate cycle to the heating cycle. Automatic change-over is needed because on some days the change-over between two cycles may take place two or three times a day. However, for installations where the primary air system is suitable, its operating economy during the summer months is usually much better than that of a straight reheating system. Hence, the added complication of the control system may be justified.

Incidentally, where under maximum load conditions the sensible heat ratio is low, say, about 60 per cent or so as in Fig. 17, a straight reheating system will be more economical to operate than a primary air system. For example, suppose that the ratio line under maximum load conditions is represented by line *1-2* of Fig. 17. Suppose for this case the ratio line will never swing to the left of *1-2* and under light load will swing to the right to some position as shown by the dotted line. If an after-mix primary air system is used, the primary air must first be cooled and dehumidified to point *3*, after which it is reheated to some point like *4*. The return air is cooled by the secondary coil to some point like *5*. The primary air in the condition represented by point *4* and the return air in the condition represented by point *5* are then mixed, and the condition of the resulting mixture is represented by point *6* on the ratio line *1-2*.

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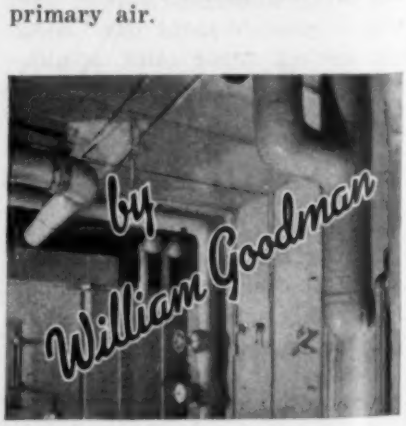
On the other hand, in a straight reheating system the air, after being dehumidified to point *3*, would simply be reheated to point *7* on ratio line *1-2*. Because a smaller quantity of air can be supplied for a straight reheating system, the quantity of reheat needed will be less than for the primary system. Thus the air is reheated only from point *3* to point *7* for the straight reheating system. For the primary air system, the air must be reheated to point *4*. It can be shown that the extra heat needed for reheating for the primary air system is equal to the extra heat that would be needed to reheat the weight of air required for a straight reheating system from point *7* to point *4*.

The fact that excess heat is needed for the primary air system is not so important in itself; its importance lies in the fact that the capacity of the refrigerating plant for the primary air system must be greater than for the straight reheating system because of the extra heat required for reheating. It should be clearly understood that the primary air system requires more refrigerating capacity than a straight reheating system only when reheat is required under maximum load conditions; that is, when the ratio line under maximum load conditions is steep, like *1-2* of Fig. 17. For almost flat ratio lines under maximum load conditions, like *1-2* of Fig. 15 [page 409, July issue], the primary air system is, of course, superior in operating economy to the straight reheating system if the ratio lines during the entire cooling season always lie to the left and above *1-8* of Fig. 15.

The primary air system is, of course, much more limited in its application than a straight reheating system because it is suitable only for those installations where the sensible heat load is always above, say, 65 per cent of the total load, and varies from 90 per cent down to, say, 65 per cent if any control range is to be obtained. As is obvious from Fig. 17, if the primary air is reheated, any control range can be obtained and the primary air system can provide satisfactory conditions for any sensible heat ratio. If the operating periods when the sensible ratio is less than about 65 per cent are relatively few in number, reheating the primary air will not noticeably affect the operating cost. The fact that the primary air system can be operated the major part of the time without reheating makes it more economical to operate than a straight reheating system. If the foregoing conditions do not exist, then a straight reheating system is the only one that will work satisfactorily for wide variations in sensible load.

Refrigerating Plants for Primary Air Systems

Before leaving the subject of primary air systems, a few words on the refrigerating equipment are needed. The primary air is, of course, cooled to a lower temperature than the secondary air. Also, the primary air is usually cooled at one central point. For this reason, direct expansion coils can be used for cooling the primary air. On the other hand, the use of chilled water is desirable for the secondary coils because these coils are usually scattered throughout a building. The water can usually be cooled in a water chiller which operates at the same suction temperature as the evaporator for the primary air.

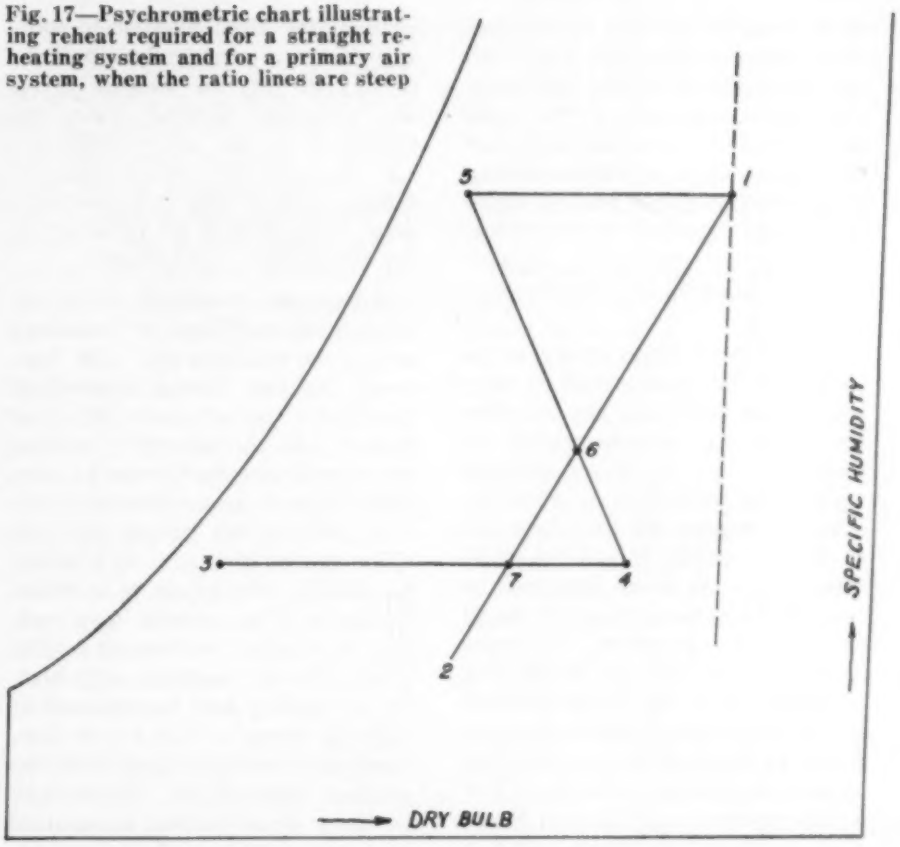


During the in-between season cycle, chilled water is supplied to the secondary coils, whereas during the winter heating cycle warm water is supplied to the secondary coils. The change-over between the two cycles may occur in some weather two or three times a day. On the heating cycle the warm water will probably be at a temperature of about 90 F while on the cooling cycle the water may be around 50 F. At the time when the change-over is made from heating to cooling, the compressor will start with 90 F water flowing through the water chiller. The suction pressure will consequently rise. If a reciprocating compressor is used, the high suction pressure due to the 90 F water flowing through the chiller will not overload the compressor motor. As was shown in the article entitled, Horsepower Requirements of Refrigerating Compressors, which appeared in the December 1940 and January and February 1941 issues of HPAC, the horsepower rises to a maximum with increasing suction pressures and then steadily decreases as the suction pressure increases still further. Inasmuch as most "Freon" compressors for air conditioning service normally operate about at suction pressures at which the maximum horsepower is required, the compressor motor will not be overloaded if the suction pressure rises during the period immediately after the control changes from the winter heating cycle to the intermediate cycle. The 90 F water can safely be allowed to flow through the chiller if a reciprocating compressor is used.

Preventing Overloading of Centrifugal Compressor Motor

However, if a centrifugal compressor is used, some means must be used to prevent the compressor motor from overloading. The horsepower of centrifugal compressors rises steadily as the suction pressure increases. Consequently, if the compressor is started when warm water at a temperature of about 90 F is circulated through the chiller, the compressor motor will be overloaded. Hence, if centrifugal compressors are used for a primary air system having an automatic change-over which requires the cooling of water initially at about 90 or 95 F, some means must be provided to prevent the suction pressure from rising high enough to overload the motor.

Fig. 17—Psychrometric chart illustrating reheat required for a straight reheating system and for a primary air system, when the ratio lines are steep



One method which can be used is illustrated in Fig. 18. A three-way valve is used to bypass the water around the chiller. The three-way valve is controlled by means of a pressure regulator actuated by the suction pressure. If the suction pressure is above some maximum predetermined point, enough of the warm water will be bypassed around the chiller to prevent the suction pressure from rising too high. As the water is cooled, the suction pressure will drop and the three-way valve will send a greater percentage of the water through the chiller. After the water has been cooled to some predetermined point, the pressure regulator will keep the bypass closed and permit all of the water to flow through the chiller. For all water temperatures and corresponding suction pressures below the setting of the pressure regulator, the bypass will, of course, remain closed so that all of the water is circulated through the chiller. Consequently there is no danger of a freeze-up due to bypassing water around the chiller.

Winter-Summer Change-Over

With almost every type of system except the straight reheating system, some method of automatically changing the functioning of the temperature control instruments is required when the system changes from summer cooling to winter heating, and vice versa. For example, in the primary air system chilled water is supplied to the secondary coils in summer and warm water in winter. During the summer, on a rise in temperature, the room thermostat must *increase* the supply of chilled water to the coil. In winter, on the other hand, on a rise in temperature the thermostat must decrease the supply of warm water to the coil. When the system changes from heating to cooling, or vice versa, the functioning of the room thermostat must also be reversed.

For year 'round systems, the change-over must be accomplished automatically. The automatic change-over is needed because, during the in-between seasons of spring and fall, changes from heating to cooling and back to heating

again occur frequently all through these seasons. Not only may cooling be required on one day while heating is required on the next, but even on the same day heat may be required in the morning, cooling in the afternoon, and heating again in the late afternoon and evening. Consequently, automatic change-over is needed for satisfactory functioning.

In buildings where there is no objection to opening windows during spring and fall, the change-over is not particularly critical. In such buildings, a direct change-over to the heating cycle when the outdoor temperature falls to about 65 F or even 60 F will be satisfactory assuming the heat can be controlled so that only a very small amount is provided in mild weather. One way of doing this, of course, is to use circulating water as the heating medium because it can be supplied at relatively low temperature in mild weather. But where windows must be kept closed the year 'round, the change-over in intermediate weather is critical and must be carefully planned and coordinated with the design of the air conditioning system.

If the air conditioning system is to be designed so that opening of windows will not be necessary during the spring and fall weather, special care must be exercised in providing means for either heating or cooling of rooms as needed, and for providing a satisfactory change-over system. A satisfactory change-over system should not change the system completely from cooling to heating or from heating to cooling. During in-between weather both heating and cooling must be available as it can be in the primary air system. In such a system, the change-over simply governs the broad functioning of the system. In this system, the change-over from heating to cooling does not completely eliminate cooling during the heating cycle, nor eliminate heating when the system is on the intermediate cycle. It is for this reason that automatic change-over at a fixed outdoor temperature is satisfactory for the primary air system.

Sometimes, however, even in windowless buildings or buildings where the windows are to be kept closed the year 'round, an outdoor thermostat set at about 60 F is used to change the entire system, or perhaps one zone, over to complete heating or to complete cooling. During the spring and fall, if the system or zone is on heating, no cooling whatsoever is possible. Similarly if the system is on cooling, no heating whatsoever is possible. Even for a single zone, heating or cooling may be required at different times of day for reasons which have nothing to do with the outdoor temperature. There may be times when heating is required even though the outdoor temperature is slightly above 60 F, and other times when cooling will be required even though the outdoor temperature is below 60 F. Whether or not cooling or heating will be required during in-between weather frequently depends upon such factors as whether the day is cloudy or bright, upon the number of people in the space, upon the electrical load, and upon the lag due to the heat storage capacity of the building and furniture. A change-over thermostat actuated only by the outdoor temperature fails to take into account the conditions inside the building when the change-over is made. For an air conditioning system which provides only heating or cooling at any one time, a change-over dependent only upon a fixed outdoor temperature is purely arbitrary and can cause considerable operating difficulty during the in-between seasons of spring and fall.

Two Thermostats in Room or Return Air

Another method sometimes used for change-over utilizes two thermostats in the room or return air, one controlling the heating equipment and the other the cooling equipment. For the proper functioning of this change-over method, the two room thermostats must be set several degrees apart. Suppose that the cooling thermostat is set to operate between 77 F and 80 F while the heating thermostat is set to operate between 70 F and 73 F. As the temperature in the conditioned room drops below 77 F, the cooling equipment is cut off and it now becomes possible for the heating equipment to run providing the temperature drops to 73 F. Once the temperature reaches 70 F, the heating equipment is turned on and continues to run until the conditioned room reaches 73 F, at which time the heating equipment will cut off. The controls are so wired that the heating and cooling equipment cannot run simultaneously, but this change-over method can work satisfactorily only if there is a fairly large temperature interval between the cutoff point of the cooling thermostat and the cut in point of the heating thermostat.

Aside from the difficulty of properly synchronizing the operation of two separate commercial thermostats, and of maintaining such synchronization over a long period of time, such a change-over system fails completely to take into account the legitimate vagaries of people. There are many who during the winter will keep a thermostat set to perhaps between 74 F and 77 F. The cooling thermostat, on the other hand, may be adjusted at the end of the season in relatively cool weather to maintain a temperature between 72 F and 75 F. Under these conditions, even though the heating and cooling equipment cannot run simulta-

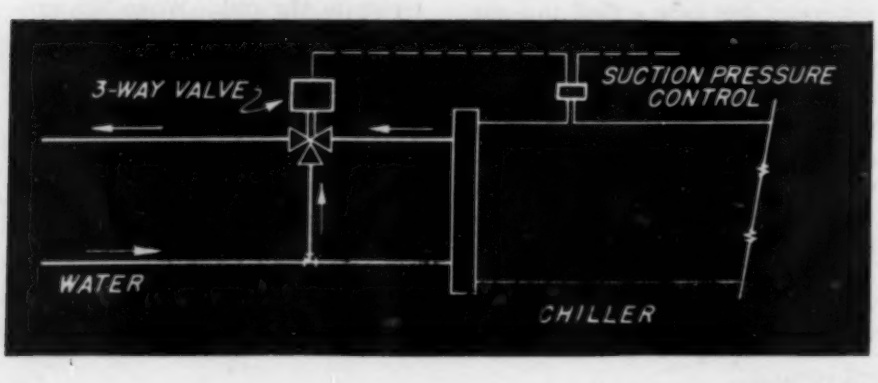


Fig. 18—Suggested method of preventing overloading of centrifugal compressors when warm water is returned to the chiller during a change from the heating to the cooling cycle

neously, they can run alternately. As soon as the heating thermostat is satisfied and stops the heating equipment, the cooling equipment will immediately start running and will run until the room has been cooled to the low point of the cooling thermostat. At this time the cooling equipment will stop and the heating equipment will immediately go on.

Any time that the ranges of the two thermostats cross each other, the heating and cooling equipment can operate alternately—a definitely unsatisfactory condition. People who own air conditioning systems with automatic control rightly feel that the equipment ought to maintain any reasonable temperatures at which they set the thermostat, and that the system should be so installed that they can maintain any one of these temperatures without crossing up the normal functioning of the equipment.

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POSTWAR PLANS

The basic principles that govern the design and control of year 'round conditioning systems are organized and discussed in the series of articles of which this is the seventh, and which is devoted to comparing primary air systems and straight reheating systems, refrigerating plants for primary air systems, and winter-summer change-over. No attempt is made in these articles to show a collection of control and equipment arrangements except insofar as they are needed to illustrate the various points discussed. Instead, the methods of accomplishing certain desired results in air conditioning systems and the broad basic principles that govern the applicability of different types of conditioning systems and controls are presented from a refreshingly new viewpoint. Those who are now designing and will be designing comfort air conditioning systems for postwar installation will find this discussion timely and practical; for no matter what the developments of the future, basic principles will always govern. Mr. Goodman, the author, is consulting engineer, the Trane Co., and a member of HPAC's board of consulting and contributing editors

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Manual Change-Over From Winter to Summer

Manual change-over from winter to summer is sometimes provided, but this is unsatisfactory in the in-between seasons because the change from heating to cooling all through the in-between seasons is likely to be both frequent and unpredictable. Manual change-over requires too much attention on the part of the operator. Perhaps manual change-over may be satisfactory for a single large air conditioned space served by a central air conditioning plant. Whether or not manual change-over will be satisfactory under these conditions depends upon the vigilance and duties of the operator, but a manual change-over system is generally not satisfactory for large systems serving many different zones.

Another method of providing winter-summer change-over is in effect the same as manual change-over although it may not always be immediately recognizable as such. Sometimes the control system changes over to the winter heating cycle whenever steam is supplied to the air heating coils. In installations where the matter of supplying steam is a manual operation dependent upon the judgment of the operator, the change-over from winter to summer is, of course, exactly the same as a straight manual change-over and suffers from exactly the same defects.

The most satisfactory air conditioning system of all is one in which no reversal of the control action is required. In a system which maintains the same action of the control equipment all through the year, there is no difficulty in maintaining satisfactory temperatures especially in the usually troublesome in-between seasons. One of the advantages of control by reheating is the fact that no reversal of the thermostat action is required and that smooth control is obtained not only in winter and summer, but in the in-between seasons when heating and cooling requirements change frequently and erratically—a point that is frequently overlooked in designing year 'round air conditioning systems.

CENTRIFUGAL PUMPS AND BLOWERS

Oddly enough, books in English covering the subject of centrifugal pumps and centrifugal blowers from a modern viewpoint have been lacking. Centrifugal Pumps and Blowers, by Austin H. Church, associate professor of machine design, New York University, is thoroughly modern in its treatment and incorporates a great deal of the most recent experimental results and data. Much material from widely different sources has been gathered and presented with clarity and thoroughness. What is even more valuable to the reader, the material is well organized and treated from an integrated point of view.

The book is written primarily for the man who buys and uses centrifugal pumps and compressors. Nevertheless, it has enough material so that it should also be of value to designers as a convenient reference work. The user of pumps, by referring to this book, can become acquainted with the latest developments and sufficiently familiar with them to discuss with confidence the meaning of a number of terms which have come into common use only in recent years.

The first two chapters cover introductory and basic material. The third chapter gives a conventional but clear presentation of the basic theory of pumps and blowers. However, more than just the usual ideal vector diagrams is covered. Such topics as the incompleteness of the ideal equations, circulatory flow, prerotation of liquid, and pulsation are discussed in enough detail to give the reader an excellent background in these topics.

Specific speed and efficiencies of pumps is presented in chapter 4; this is a topic that is usually barely mentioned in most treatments. This chapter treats the relationship of specific speed and efficiency in enough detail to give an excellent idea of the various types of pumps that are best suited for different applications. Specific speed is a basic coordinate and the

[*Concluded on page* 529]

Answering the Question:

HOW MUCH OF MY FUEL SHOULD I HAVE USED?

ON PAGES 410-412 of the August 1943 HPAC and page 464 of the September 1943 HPAC, we published data showing the percentages of fuel used in the various months of the heating season for a number of cities in the United States and Canada.

Two tables were published. Table 1 showed the percentage of the total fuel burned in each month for a year of normal degree days. Table 2 showed the total percentage of fuel burned from October 1 to the end of any given month.

The total percentages in Table 2 were accumulated from October 1 because at that time fuel oil rationing regulations were based on a heating season beginning with October 1. The regulations are now based on a heating season beginning with September 1. Hence, for the convenience of our readers, Table 2 has been recomputed. The revised table showing the total percentages of fuel burned from September 1 to the end of any given month is presented herewith.

The table is based on data obtained from Weather Bureau records for a number of years. However, it is quite possible in any locality for the degree days in any one year to be greater or lower than the normal. In addition, because some months may be warmer or colder than normal, it is possible for the monthly degree days for any one year to differ somewhat from the normal degree days for that month. Nevertheless, the percentages given can be of considerable help in roughly estimating whether any particular building is burning more or less fuel than it should according to the total allotted to it under rationing regulations.

The data apply to a building normally heated to 70 F indoor temperature, inasmuch as the degree day figures from which the percentages are computed are based on a 70 F indoor temperature.

Accumulated percentages from September 1 to end of month. These accumulated percentages show total fuel burned from September 1 to end of any month for a year of normal degree days

	Sept.	Oct.	Nov.	Dec.	Jan.	Feb.	Mar.	Apr.	May	June	July	Aug.
ALABAMA												
Birmingham	...	2.1	15.9	39.8	65.4	85.2	97.6	100.0	...	...	...	...
Mobile	...	0.2	13.2	40.2	68.6	88.2	99.3	100.0	...	...	...	...
ARIZONA												
Phoenix	...	...	11.4	40.1	70.5	90.2	99.7	100.0	...	...	...	...
ARKANSAS												
Fort Smith	...	2.9	16.4	39.2	64.6	84.6	96.9	100.0	...	...	...	...
Little Rock	...	2.5	16.0	38.5	64.1	83.8	96.8	100.0	...	...	...	...
CALIFORNIA												
Los Angeles	...	0.7	9.0	26.7	48.6	66.7	82.5	92.9	99.8	100.0	...	...
San Francisco	3.8	8.1	16.2	29.3	43.7	54.7	65.0	74.3	82.1	88.1	94.3	100.0
COLORADO												
Denver	1.3	8.6	21.4	38.6	57.1	72.5	86.0	95.1	99.7	100.0	...	...
Grand Junction	0.5	7.2	20.8	41.2	63.6	79.4	91.1	97.8	100.0	...	...	...
CONNECTICUT												
New Haven	0.9	6.8	18.5	35.5	54.8	71.8	87.1	96.1	99.8	100.0	...	...
DIST. OF COLUMBIA												
Washington	0.1	5.2	18.0	37.0	58.2	76.2	91.2	98.7	100.0	...	...	...
FLORIDA												
Jacksonville	...	...	9.5	38.6	70.7	91.8	100.0	...	...	...	...	...
GEORGIA												
Atlanta	...	2.8	16.3	38.2	62.4	81.7	95.8	100.0	...	...	...	...
Savannah	...	0.4	13.2	38.9	66.6	86.8	99.0	100.0	...	...	...	...
IDAHO												
Boise	1.8	9.5	22.3	40.5	59.9	75.0	87.3	95.1	99.4	100.0	...	...
ILLINOIS												
Chicago	0.3	5.4	17.2	35.2	55.7	73.2	88.0	96.6	99.9	100.0	...	...
Springfield	0.1	5.4	18.1	37.2	59.3	77.7	91.9	98.7	100.0	...	...	...
INDIANA												
Evansville	...	4.1	17.2	37.6	60.7	79.7	93.7	99.6	100.0	...	...	...
Indianapolis	0.2	5.6	18.4	37.5	58.8	76.6	91.2	98.5	100.0	...	...	...
IOWA												
Des Moines	0.4	6.0	18.5	37.4	59.1	77.1	91.2	98.2	100.0	...	...	...
Sioux City	1.0	7.2	19.9	38.3	59.0	76.5	90.7	98.0	100.0	...	...	...
KANSAS												
Dodge City	0.1	5.5	18.8	38.6	60.7	78.3	91.9	98.7	100.0	...	...	...
Topeka	0.1	5.0	18.0	38.0	60.7	79.4	93.0	99.0	100.0	...	...	...
KENTUCKY												
Lexington	...	5.1	18.3	38.0	59.6	77.6	92.0	99.0	100.0	...	...	...
Louisville	...	4.2	17.3	37.6	60.3	78.9	93.4	99.6	100.0	...	...	...
LOUISIANA												
New Orleans	...	...	10.2	38.8	71.8	93.0	100.0	...	...	...	...	...
Shreveport	...	1.2	15.0	40.1	68.5	88.6	99.2	100.0	...	...	...	...
MAINE												
Eastport	3.3	9.7	19.7	33.9	50.2	64.6	77.8	87.0	93.3	96.8	98.5	100.0
Portland	2.2	8.7	19.9	36.6	54.3	70.3	84.6	93.8	98.8	99.9	...	100.0
MARYLAND												
Baltimore	0.1	4.8	17.2	36.2	57.6	75.9	91.5	99.1	100.0	...	...	...
MASSACHUSETTS												
Boston	1.0	6.9	18.4	35.2	54.3	71.2	86.4	95.7	99.8	100.0	...	...
MICHIGAN												
Detroit	0.9	6.9	18.8	35.9	55.4	72.6	87.8	96.5	99.9	100.0	...	...
Marquette	2.6	9.1	20.0	35.1	52.4	68.0	82.3	91.7	97.4	99.5	99.6	100.0
MINNESOTA												
Duluth	3.0	9.6	20.3	35.8	53.9	69.2	82.3	90.9	96.5	98.9	99.2	100.0
Minneapolis	1.4	7.7	20.1	38.0	58.6	76.0	89.9	97.0	99.9	100.0	...	...
MISSISSIPPI												
Vicksburg	...	1.2	14.8	39.9	68.1	88.1	99.0	100.0	...	...	...	...
MISSOURI												
Kansas City	...	4.5	17.3	37.5	60.3	79.2	93.0	99.1	100.0	...	...	...
St. Louis	...	4.2	17.2	37.8	60.9	79.5	93.8	99.7	100.0	...	...	...
Springfield	...	4.8	17.9	38.1	60.2	79.1	93.0	99.1	100.0	...	...	...
MONTANA												
Havre	3.0	10.4	22.1	38.1	56.8	73.5	87.1	94.5	98.7	99.8	...	100.0
NEBRASKA												
Lincoln	0.3	5.8	18.6	37.8	59.4	77.4	91.5	98.2	100.0	...	...	...
Omaha	0.2	5.6	18.5	38.0	59.7	77.7	91.8	98.5	100.0	...	...	...
NEVADA												
Winnemucca	2.8	11.0	23.6	40.7	58.5	72.4	84.7	93.4	98.8	100.0	...	...
NEW HAMPSHIRE												
Concord	2.4	8.9	20.1	36.3	54.8	71.0	85.6	94.5	99.1	100.0	...	...
NEW JERSEY												
Atlantic City	0.2	5.0	16.2	33.4	52.9	69.9	85.7	95.7	99.8	100.0	...	...
NEW MEXICO												
Santa Fe	2.0	9.4	22.3	39.8	58.2	72.9	85.8	94.8	99.5	100.0	...	...
NEW YORK												
Albany	1.0	7.1	18.9	36.2	56.1	73.6	88.9	97.3	100.0	...	...	...
Buffalo	1.2	7.1	18.4	34.4	52.8	69.5	84.9	94.7	99.4	100.0	...	...
New York	0.2	5.3	17.1	34.7	54.7	72.5	88.5	97.4	100.0	...	...	...

FUEL EFFICIENCY COORDINATORS ADDED

A listing of the coordinators appointed to conduct the voluntary national fuel efficiency program in cooperation with the U. S. Bureau of Mines was published on pages 400 and 401 of the July HPAC, and on pages 65 and 66 in the front section of the August issue.

The names of additional coordinators since appointed follow:

California

Bakersfield Area—John J. Wilt, 2114 B St., Bakersfield.

Illinois

Danville-Urbana Area — Prof. Julian R. Fellows, University of Illinois, Urbana.

Galesburg Area—Ira E. Neifert, Knox College, Galesburg.

Mt. Carmel-Fairfield Area—C. W. McCullough, 500 E. Delaware St., Fairfield.

Massachusetts

Pittsfield Area—Weston Morrill, 100 Woodlawn Ave., Pittsfield.

Michigan

Houghton Area—Prof. James Fisher, Michigan College of Mining & Technology, Houghton.

Traverse City Area—Charles A. Brown, P. O. Box 306, Elk Rapids.

Montana

Butte Area—T. J. Sullivan, P. O. Box 368, Butte.

New York

Syracuse Area—John B. Foley, Jr., 249 Erie Blvd., West, Syracuse.

North Carolina

Greensboro Area—Frank E. P. Klages, 1034 Jefferson Standard Bldg., Greensboro.

Ohio

Hamilton-Lebanon Area—L. F. Wertz, Town Hall, Lebanon.

Oklahoma

Lawton Area—W. Schumacher, Box 458, Lawton.

Pennsylvania

Pottsville Area—E. A. Reilly, Philadelphia & Reading Coal & Iron Co., Pottsville.

Scranton Area—C. Schillinger, 1109-1114 Markle Bank Bldg., Hazleton.

York Area—H. A. Delano, American Chain & Cable Co., Inc., York.

Tennessee

Nashville Area—L. C. Peal, Bureau of Inspection & Permits, Nashville.

Texas

San Antonio Area—Lloyd D. Royer, 911 Smith-Young Tower, San Antonio.

Wichita Falls Area—William H. Rouzer, Jr., City National Bldg., Wichita Falls.

West Virginia

Huntington Area—E. J. Kerr, Guaranty Bank Bldg., Huntington.

Steubenville-Weirton Area — F. C. Sweeney, Weirton Steel Co., Weirton.

Continued—Accumulated percentages from September 1 to end of month. These accumulated percentages show total fuel burned from September 1 to end of any month for a year of normal degree days

	Sept.	Oct.	Nov.	Dec.	Jan.	Feb.	Mar.	Apr.	May	June	July	Aug.
NORTH CAROLINA												
Raleigh	...	3.1	16.3	37.8	61.1	80.3	94.7	100.0	...	...	...	...
Wilmington	...	1.3	13.0	34.4	59.3	80.1	95.9	100.0	...	...	...	...
NORTH DAKOTA												
Bismarck	2.3	9.1	21.1	38.2	57.6	74.4	88.3	95.8	99.4	100.0	...	...
OHIO												
Cincinnati	0.2	5.8	19.0	38.1	59.1	76.7	91.2	98.6	100.0	...	...	...
Cleveland	0.8	6.5	18.3	35.3	54.7	71.8	87.1	96.3	99.9	100.0	...	...
Columbus	0.3	5.9	18.7	37.4	58.2	75.9	90.7	98.3	100.0	...	...	...
OKLAHOMA												
Oklahoma City	...	3.3	16.7	38.7	63.2	82.8	95.6	99.9	100.0	...	...	...
OREGON												
Baker	3.7	11.6	23.7	39.9	57.1	71.1	82.8	91.0	96.7	99.4	99.6	100.0
Portland	2.3	9.8	22.1	38.7	56.6	71.0	83.6	92.5	98.2	100.0	...	...
PENNSYLVANIA												
Philadelphia	0.1	4.8	16.9	35.5	56.5	74.7	90.3	98.4	100.0	...	...	...
Pittsburgh	0.3	5.9	18.5	36.9	57.4	75.1	90.3	98.3	100.0	...	...	...
SOUTH CAROLINA												
Charleston	...	0.5	12.5	36.4	63.6	84.1	97.8	100.0	...	...	...	...
Columbia	...	2.3	16.3	39.6	64.5	84.4	97.3	100.0	...	...	...	...
SOUTH DAKOTA												
Huron	1.4	8.0	20.3	37.8	58.2	75.6	89.2	96.5	99.8	100.0	...	...
Rapid City	1.9	9.0	21.1	37.5	56.0	72.1	86.0	94.6	99.3	100.0	...	...
TENNESSEE												
Knoxville	...	4.4	18.6	39.7	62.1	80.0	94.0	99.8	100.0	...	...	...
Memphis	...	2.6	16.1	38.5	63.8	83.4	96.7	100.0	...	...	...	...
Nashville	...	3.7	17.4	38.7	62.1	80.8	94.8	99.9	100.0	...	...	...
TEXAS												
El Paso	...	2.9	17.8	43.0	68.0	86.1	97.6	100.0	...	...	...	...
Fort Worth	...	1.1	14.2	39.1	67.0	88.5	98.9	100.0	...	...	...	...
Houston	...	...	10.7	39.5	72.8	95.1	100.0	...	...	...	...	...
San Antonio	...	...	11.3	40.0	72.3	94.3	100.0	...	...	...	...	...
UTAH												
Modena	2.3	10.3	23.3	40.1	58.1	72.5	85.1	93.7	99.1	100.0	...	...
Salt Lake City	1.0	7.9	20.7	39.0	58.8	74.4	87.3	95.6	99.8	100.0	...	...
VERMONT												
Burlington	1.9	8.4	19.9	36.7	55.7	72.7	87.5	96.2	99.7	100.0	...	...
VIRGINIA												
Lynchburg	...	5.2	19.1	39.6	61.7	79.6	93.8	99.8	100.0	...	...	...
Norfolk	...	2.7	14.9	35.2	57.8	76.5	92.1	99.4	100.0	...	...	...
Richmond	...	4.4	17.5	38.0	60.0	78.6	93.0	99.6	100.0	...	...	...
WASHINGTON												
Seattle	4.0	12.3	23.7	37.8	53.3	66.4	78.6	87.7	94.1	97.6	98.8	100.0
Spokane	2.8	11.0	23.6	40.6	59.0	74.0	86.4	94.3	98.9	100.0	...	...
WEST VIRGINIA												
Elkins	1.2	8.2	21.3	39.0	58.0	74.5	88.2	96.8	100.0	...	...	...
Parkersburg	0.2	5.9	19.2	38.4	59.4	77.3	91.6	98.8	100.0	...	...	...
WISCONSIN												
Green Bay	1.8	8.3	20.1	36.8	56.2	73.1	87.4	95.6	99.6	100.0	...	...
La Crosse	1.2	7.4	19.7	37.8	58.5	76.1	90.3	97.6	100.0	...	...	...
Milwaukee	1.1	7.1	18.7	35.6	54.9	71.4	85.7	94.6	99.3	100.0	...	...
WYOMING												
Cheyenne	3.2	11.5	23.6	38.7	55.0	69.1	82.3	91.9	98.0	99.8	100.0	...
Lander	3.4	11.4	24.0	40.7	58.2	72.6	84.8	93.0	98.2	99.8	...	100.0
CANADA												
ALBERTA												
Calgary	4.5	12.0	23.8	37.8	54.7	69.1	81.6	89.2	94.2	96.9	98.1	100.0
Edmonton	4.4	11.3	23.3	38.1	55.9	70.6	83.2	90.2	94.4	97.0	98.2	100.0
BRITISH COLUMBIA												
Vancouver	4.7	13.3	24.8	39.3	54.9	68.1	80.5	89.4	95.3	98.4	99.5	100.0
MANITOBA												
Winnipeg	2.4	9.1	20.9	37.3	56.5	72.9	87.1	94.4	98.6	99.4	...	100.0
NEW BRUNSWICK												
Moncton	3.4	10.4	20.8	35.8	52.9	69.0	82.3	91.4	96.6	99.0	...	100.0
NOVA SCOTIA												
Halifax	2.7	9.2	19.3	34.2	51.2	66.5	80.6	90.8	97.3	100.0	...	...
ONTARIO												
Ottawa	2.4	9.2	20.6	37.4	56.7	73.8	88.4	96.4	99.6	100.0	...	...
Port Arthur	3.4	10.2	21.1	35.9	53.3	68.8	82.4	91.0	96.3	98.6	99.2	100.0
Toronto	2.3	9.5	20.8	36.5	53.8	69.4	85.1	94.4	99.2	100.0	...	...
PRINCE EDWARD IS.												
Charlottetown	7.2	13.9	24.2	39.0	53.0	66.4	80.8	91.2	97.5	100.0	...	...
QUEBEC												
Montreal	2.2	8.9	20.4	37.1	56.1	73.2	87.7	96.3	100.0	...	...	...
Quebec	2.8	9.7	20.8	36.8	54.8	70.5	84.6	93.8	98.4	99.7	...	100.0
SASKATCHEWAN												
Saskatoon	3.9	10.9	22.1	37.2	55.6	71.5	85.3	92.3	96.3	98.1	98.6	100.0

AIR CONDITIONING INCREASES PRODUCTION, IMPROVES QUALITY

E. Rasmussen Discusses Installation for Circuit Breaker Assembly Room Which Holds Constant Temperature of 77 Degrees the Year Around and Replaces Separate Testing Enclosure Formerly Used

AIR CONDITIONING has many applications in industry, but seldom is it necessary to control the temperature within such close limits as is done in the circuit breaker assembly room at the East Pittsburgh plant of Westinghouse.

Because the tripping mechanism of small circuit breakers responds to heat as generated by the flow of electric current, and because of the small deflection of these elements on breakers rated up to 100 amp at 600 volts, each unit must be accurately calibrated at the same uniform temperature. Previous to the application of temperature control to the entire assembly area, the breakers were tested in a separate air conditioned enclosure. This necessitated removing the units from the assembly line, stacking them in the enclosure, and allowing them to stand until they reached the temperature of the test room, which was kept at 77 F. Now the checking is merely the final step in the production process. Furthermore, since the temperature is held to within plus or minus 1 F, the uniformity of the product has been improved.

Temperature in the assembly room, which is 60 ft wide, 240 ft long, and has 16 ft ceilings, is held at 77 F DB $\pm$ 1 F the year 'round. With the exception of removing all windows and replacing them with glass brick, few changes were made in the room. Lighting equipment remained as it was—partly incandescent and partly fluorescent—and it was not necessary to insulate the walls, which are brick. Fig. 1, a general view of the work area, shows the sections that were replaced with glass brick and illustrates the size of the assembly room.

Fig. 1—The entire small circuit breaker assembly room, with an area of nearly 15,000 sq ft, is kept within 1 deg of 77 F dry bulb the year around. Air conditioning has proved its worth by materially increasing production and improving quality

Since the amount of moisture in the air does not affect the product, the humidity is not controlled during the summer months. However, in the winter, it is kept to within 30 to 40 per cent for considerations of health and comfort. All air conditioning equipment was installed adjacent to and on the same level with the assembly room.

The cooling load was calculated

as 1,126,000 Btu with an outside air temperature of 100 F DB and 80 F WB, and an inside temperature of 77 F DB with 50 per cent relative humidity. The heating load was calculated as 1,000,000 Btu with an outside temperature of 0 F and 77 F inside temperature. The internal load was calculated for 200 people, with 43 kw for lighting and 45 kw for electrical appliances, motors, etc.

Industrial applications of air conditioning for increasing production, improving the quality of the product, and promoting the health and comfort of workers are many and are bound to increase in the competitive postwar period. Here's an example of a product —circuit breakers—which must be accurately calibrated at uniform temperature. Formerly, the breakers were tested in a separate air conditioned enclosure, which necessitated removing the units from the assembly line, stacking them in the enclosure, and allowing them to stand until they had reached the temperature of the space. Now, with the assembly room completely air conditioned and held to within plus or minus one degree of 77 F the year around, the checking is merely the final step in the production process and both production and quality have been improved . . . The author is works heating and ventilating engineer, Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

Two 60 Hp Compressors

The refrigerating equipment consists of two vertical, eight cylinder, single acting, direct connected, hermetically sealed, "Freon 12" compressors, which are direct connected to 60 hp, 1750 rpm, 60 cycle, 220 volt, 3 phase, totally enclosed, refrigerant cooled motors. One compressor is equipped with a variable capacity attachment. There are two evaporative condensers, each having an air capacity of 14,000 cfm. The fans are driven through a belt drive by a 5 hp, 220 volt, 60 cycle, 3 phase, 1750 rpm motor. Fan speed is 490 rpm. Condenser coil sections have a face area of 22.3 sq ft each, and water is circulated by two pumps, each having a capacity of 80 gpm at a 15 ft head. The pumps are direct connected to ¾ hp motors. Two liquid receivers each with a capacity of 368 lb of "Freon" (at 100 F), are installed between the condenser coils and the subcooling coils. These subcooling coils, installed below the condenser coils, cool the liquid refrigerant leaving the receiver to at least 5 F below condensing temperature. The evaporative condensers and circulating pumps are shown in Fig. 2.

The evaporator is made up of three steel coils having a total face area of 61.5 sq ft, five rows deep. Each coil has two distributor clusters and two suction manifolds, one for two rows and the other for the three rows of coil. Each evaporator coil has two expansion valves. Fig. 3, showing both compressors, also shows the variable capacity attachment on the No. 1 compressor.

The heating equipment consists of three reheat coils with a total face area of 60 sq ft, one row deep, and one preheat coil with a face area of 13.75 sq ft, one row deep. Forty-two humidifier spray nozzles are used to control humidity.

The air distribution equipment consists of a fan with a capacity of 42,500 cfm, of which 35,500 cfm is normally recirculated and 7000 cfm is outside air. The 7000 cfm is the amount of air exhausted through various ventilating and dust collecting systems in the assembly room. The fan is driven through a v belt driven by a 20 hp, 220 volt, 60 cycle, 3 phase, 1750 rpm motor.

Air is distributed through two main ducts the full length of the room. Due to wartime scarcities, the ducts were made of pressed fiber board. Sides, tops, and bottoms of the ducts are joined together by quarter-round wooden corner strips held in place by wood screws. The only metal used is in the angle iron hangers and supports throughout the entire length of the ducts.

Twenty-four high velocity air diffusers are mounted along the length of the ducts which gradually reduce in dimensions from 56 in. x 36 in. at the entrance to 24 in. x 10 in. at the far end of the room. These diffusers give complete draftless air distribution over the entire room. The return air registers are at the end of the room next to the air conditioning equipment room.

One other item of interest is the arrangement of the preheater. Since only one section of the outside air intake is covered with the

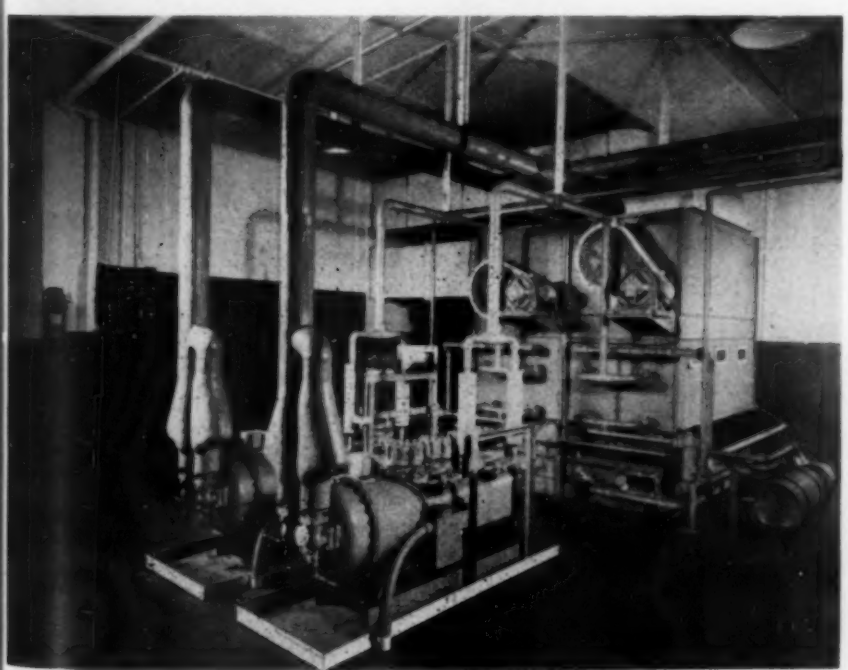
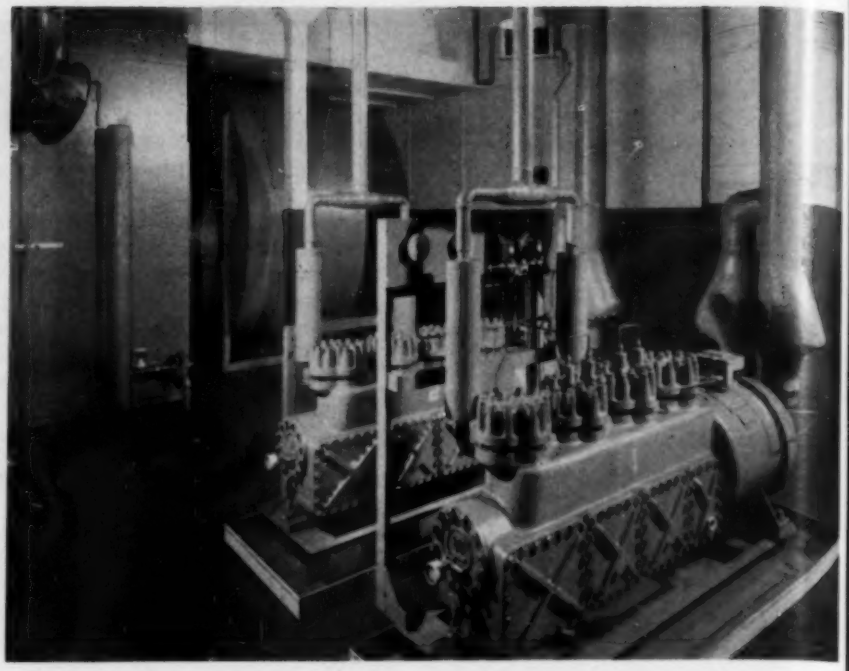


Fig. 2—The evaporative condensers are shown in back of the two compressors. Each is connected to one compressor, has 14,000 cfm capacity

Fig. 3—Mounted on top of the middle cylinder heads of compressor No. 1 are the variable capacity attachments which are controlled through a series of mercury switches by the master thermostatic control in breaker room

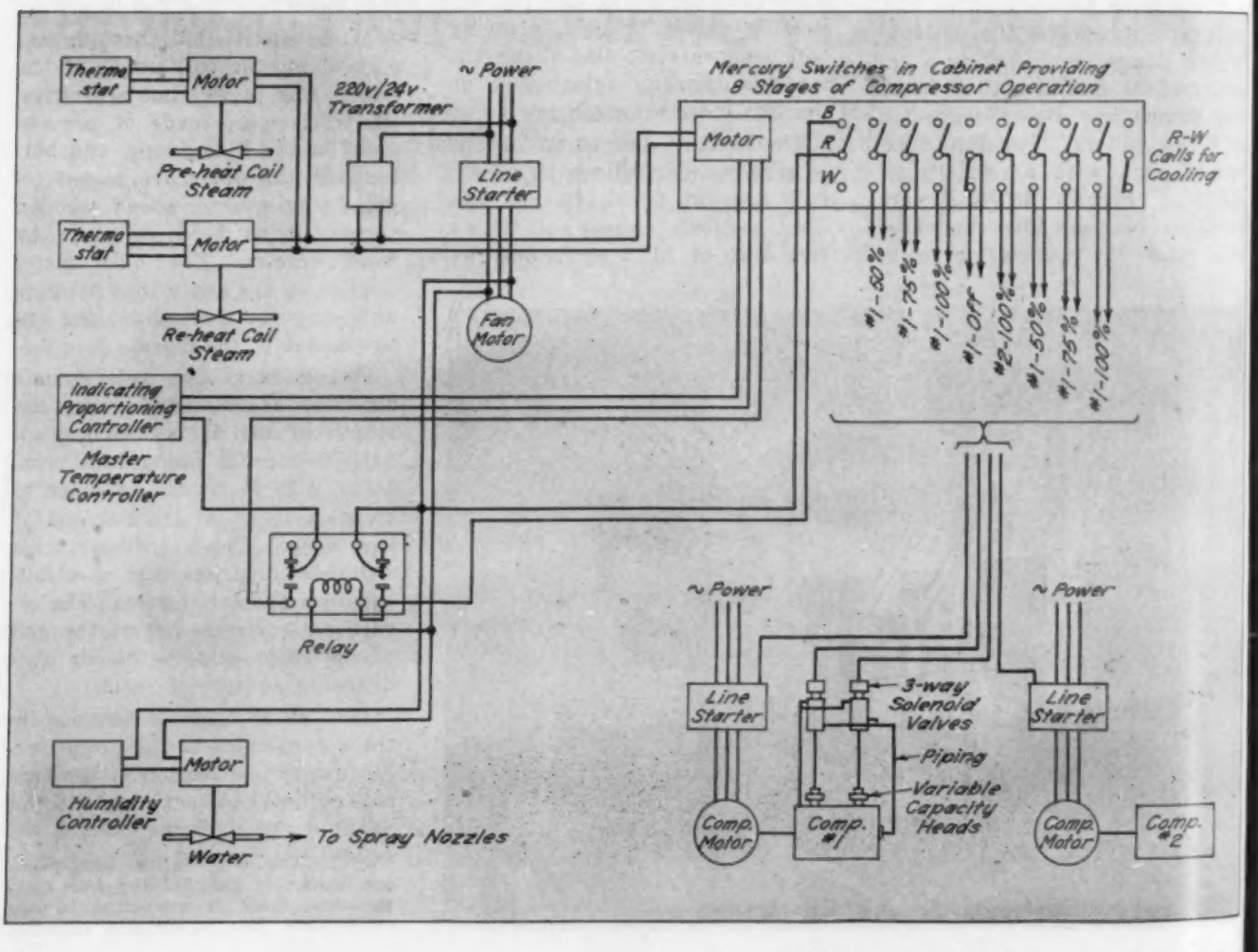


preheater coil, the maximum amount of air that can be preheated is 7000 cfm. However, when (in the spring and fall) outside conditions are such that by increasing the amount of outside air, the load can be decreased on the refrigerating system, an adjustable damper allows this to be done.

The refrigerating equipment is piped as two independent systems; that is No. 1 compressor with the variable capacity attachment is connected to No. 1 evaporative condenser and two rows of the evaporator coils. No. 2 compressor is piped to No. 2 evaporative condenser and three rows of the evaporator coils. This was done in order to secure a system as flexible as possible, and, in case of a breakdown, one system under not too severe weather conditions should be able to carry the cooling load and eliminate interruption in production in the assembly room.

As can well be imagined, rather elaborate controls are required to hold the temperature to ± 1 F both winter and summer. The circulating fan motor is interconnected, through a relay, with the main compressor controls. If anything should happen to shut down the fan, the refrigerating equip-

Fig. 4—The eight stages of compressor operation, which can be obtained with the two compressors, are controlled through a series of mercury switches by a proportioning controller



ment automatically goes off the line. Thus, if the circulating system is out of order, the thermostat in the assembly room calling for more cooling has no effect and there is no danger of the evaporator coils freezing up.

The thermostat for the main steam coils and the humidistat are connected in the conventional manner. All are, of course, located together on one panel in the assembly room. The steam preheater control functions only when the temperature of the outside air has dropped to 35 F.

Of chief interest, however, are the master and supplementary controls for the refrigerating compressors. An indicating proportioning potentiometer control regulates the temperature from a position in the assembly room. This potentiometer is connected through a series of mercury switches to both compressors.

On compressor No. 1, the variable capacity attachment, which operates from condenser pressure acting on a three-way solenoid valve, controls the working of a special cylinder head. In one position, the attachment holds open the suction valves, the refrigerant is sucked in, and is then pushed back into the intake chamber. In the other position, the suction valve is allowed to close, the refrigerant is compressed to the upper pressure limit, and the cylinder operates at full capacity.

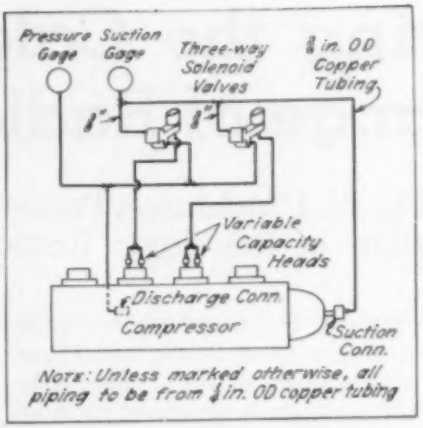


Fig. 5—The variable capacity attachment on compressor No. 1 gives from 50 to 100 per cent regulation by 25 per cent increments. When two heads are cut out, "Freon" refrigerant is pushed back into the suction chamber

Functioning of these mercury switches can probably best be described by a supposition. Suppose that, as so often happens in the spring and fall, the weather suddenly turns warm. The main thermostatic control calls for cooling. Compressor No. 1 comes on the line at 50 per cent capacity; i.e., four cylinders are operating and four are cut out. Still more cooling is required, and two more cylinders are cut in, bringing the capacity to 75 per cent. The next step brings the compressor to its full capacity.

Should more cooling be required, No. 2 compressor comes on full and No. 1 shuts down. In progressive steps, compressor No. 1 comes on again at 50, 75, and finally 100 per cent capacity. A cutting back in cooling effect is accomplished in the reverse sequence. Because of the gradual steps in the cooling cycle, this control system holds the temperature much closer than one which shuts down the compressors altogether.

Not only has this installation proved its worth by a material increase in production, but it has also helped, by holding the temperature constant, to improve the quality of the product.

[*Concluded from page* 523]

author is to be commended for giving sufficient space to it.

Chapter 5 covers the performance curves and cavitation of pumps. Cavitation has in recent years been given considerable attention; it is of importance not only because of its erosive effect on the impeller, but also because of its effect on the operation of pumps. Chilled water pumps for steam jet water cooling systems and tail pumps for steam condensers must be carefully selected in the light of our knowledge of cavitation if operating difficulties are to be avoided. This chapter covers the effect of cavitation on operation in sufficient detail so that pumps for handling water under high vacuums can be properly selected.

Chapter 6 deals with design of vanes, casings, and diffusers. Chapter 7 covers the various types of impellers and their application.

Chapters 8, 9, and 10 will be of considerable interest to users and buyers of pumps. Chapter 8 covers pump details and materials. It shows the various types of shafts and sleeves, packing boxes, casings, wearing rings, and so forth—all details in which users of pumps are likely to be highly interested. Pump applications and selection are covered in chapter 9. Chapter 10 offers pointers on installation, operation, and testing.

Chapters 11 to 17 deal altogether with centrifugal blowers. Basic thermodynamic data on the compression of gases are covered in chapter 11. Chapter 12 deals with the classification and performance curves of blowers. The design of a radial type blower is covered in chapter 13.

Chapters 14 to 17 are likely to be of greatest interest to purchasers of blowers. The construction details of modern blowers are covered in chapter 14. The applications of the various types of blowers are covered in chapter 15. Chapter 16 deals with the little understood subject of regulation of blowers and is thoroughly modern in its approach and treatment. Chapter 17 deals with installation, operation, and testing.

Chapters 18 and 19 are likely to be of interest only to designers of pumps and blowers, chapter 18 covering disk stresses and chapter 19 critical speeds.

This 308-page book contains a wealth of information presented in a clear fashion. Mathematics can hardly be avoided in a book such as this, but the author is to be commended for having kept it relatively simple. The book (which costs $4.50) is thoroughly worthwhile and will probably serve as a reference in this field for a long time to come.

The publisher is John Wiley & Sons, Inc., 440 Fourth Ave., New York 16, N. Y.—*W. G.*

NEW REPORT REQUIRED OF HEATING EQUIPMENT MAKERS: Manufacturers of extended surface heating equipment are required to make a new report on models and sizes of equipment they wish to produce, the War Production Board reported August 23.

Simplifying the Calculations for Exchange of Radiant Heat

B. F. Raber and F. W. Hutchinson Present Chart for the Graphical Solution of the Basic Radiation Equation

Because of the wide interest in radiant and panel heating methods, there has been considerable discussion in recent months of simplifying the calculations now necessary in solving problems of radiant exchange. The authors (who are professor of mechanical engineering and chairman of the department, and associate professor of mechanical engineering, respectively, at the University of California) have developed the method given here. The fundamental formula, equations for various cases, and an example showing the use of the method, together with tables of data on emissivity and shape factor, are presented with the graph

DETERMINATION OF the net radiant exchange between two surfaces separated by a non-absorbing medium is frequently both complex and tedious. The complexity arises from the need of evaluating the shape factor of one surface with respect to the other, while the tediousness is due to the need of working with the fourth power of the absolute temperature. The latter difficulty can be readily overcome, without loss in accuracy, by setting up a graphical solution of the basic radiation equation; the former difficulty cannot be avoided except through use of some form of approximation, but for a very large number of practical exchange problems such methods give results satisfactory within the limits of required engineering accuracy. The intent of this article is to present, in simple form, one approximate method of evaluating shape factors for the general case and to provide a graph obviating the need of any calculations in solving the basic equation.

The fundamental radiation formula, frequently called the Stefan-Boltzmann equation, is

$$Q_{sr} = 0.173\, F_s\, F_e \left[\left(\frac{T_s}{100} \right)^4 - \left(\frac{T_r}{100} \right)^4 \right] \quad \ldots\ldots[1]$$

where Q_{sr}=net radiant transfer from source to receiver, Btu per hr per sq ft of source; T_s, T_r=absolute temperature, F+460, of source and receiver respectively; and F_e, F_s=factors dependent on the relative emissivity and geometrical arrangement, respectively, of the surfaces between which radiant exchange is occurring. Strictly, the influence of emissivity and geometry cannot be separated into independent factors, but a combined term F_{es} should be used; in practice convenience dictates the separation.

The emissivity factor, F_e, can be expressed in terms of the emissivities, e_s and e_r, of the source and receiver. For most problems which commonly occur in practice, F_e can be evaluated by one of three equations, or can be estimated by interpolating between calculated values for two of the same three equations. These are:

Equation I: Where one surface, either source or receiver, is completely enclosed by the other surface and is small with respect to it,

$$F_e = \text{the emissivity of the enclosed surface} \quad \ldots\ldots[I]$$

Equation II: When one surface is completely enclosed by the other and both surfaces are of the same order of magnitude (a special example of this case occurs when the surfaces are infinite parallel planes.)

$$F_e = \frac{1}{\frac{1}{e_s} + \frac{1}{e_r} - 1} \quad \ldots\ldots[II]$$

Equation III: When neither surface encloses the other and the receiver intercepts only a small fraction of energy leaving the source,

$$F_e = e_s\, e_r \quad \ldots\ldots[III]$$

The shape factor, F_s, is numerically equal to the fraction of energy intercepted by the receiver of that leaving the source. Referring to Fig. 1 and considering that the areas A_s and A_r are in any position with respect to one another, it is possible to show that

$$F_s = \frac{1}{A_s} \iint \frac{\cos \phi_s\, dA_s\, dw}{\pi} \quad \ldots\ldots[2]$$

where dw is a small solid angle subtended by the infinitesimal area dA_r from a point on A_s.

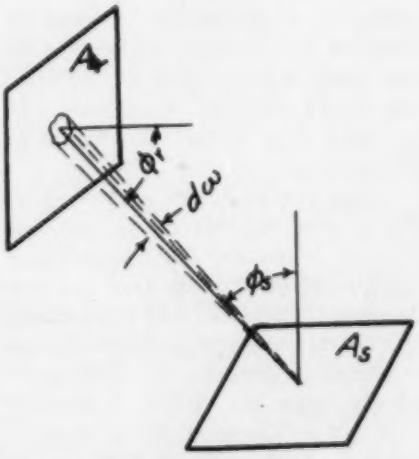


Fig. 1—Illustrating the geometry of the shape factor

But a solid angle is equal to the intercepting area projected on a plane normal to a line from the source divided by the length of that line squared,

$$dw = \frac{\cos \phi_r\, dA_r}{r^2} \quad \ldots\ldots[3]$$

or

$$F_s = \frac{1}{A_s} \iint \frac{\cos \phi_s \cos \phi_r\, dA_s\, dA_r}{\pi r^2}$$

Except in the simplest cases, equation 3 presents formidable difficulties in the numerical evaluation of shape factors. To avoid such difficulties and arrive at solutions which, though approximate, retain the rational method and in most practical problems are within the limit of permissible error, the integrations can be eliminated and the shape factor equation written in the form

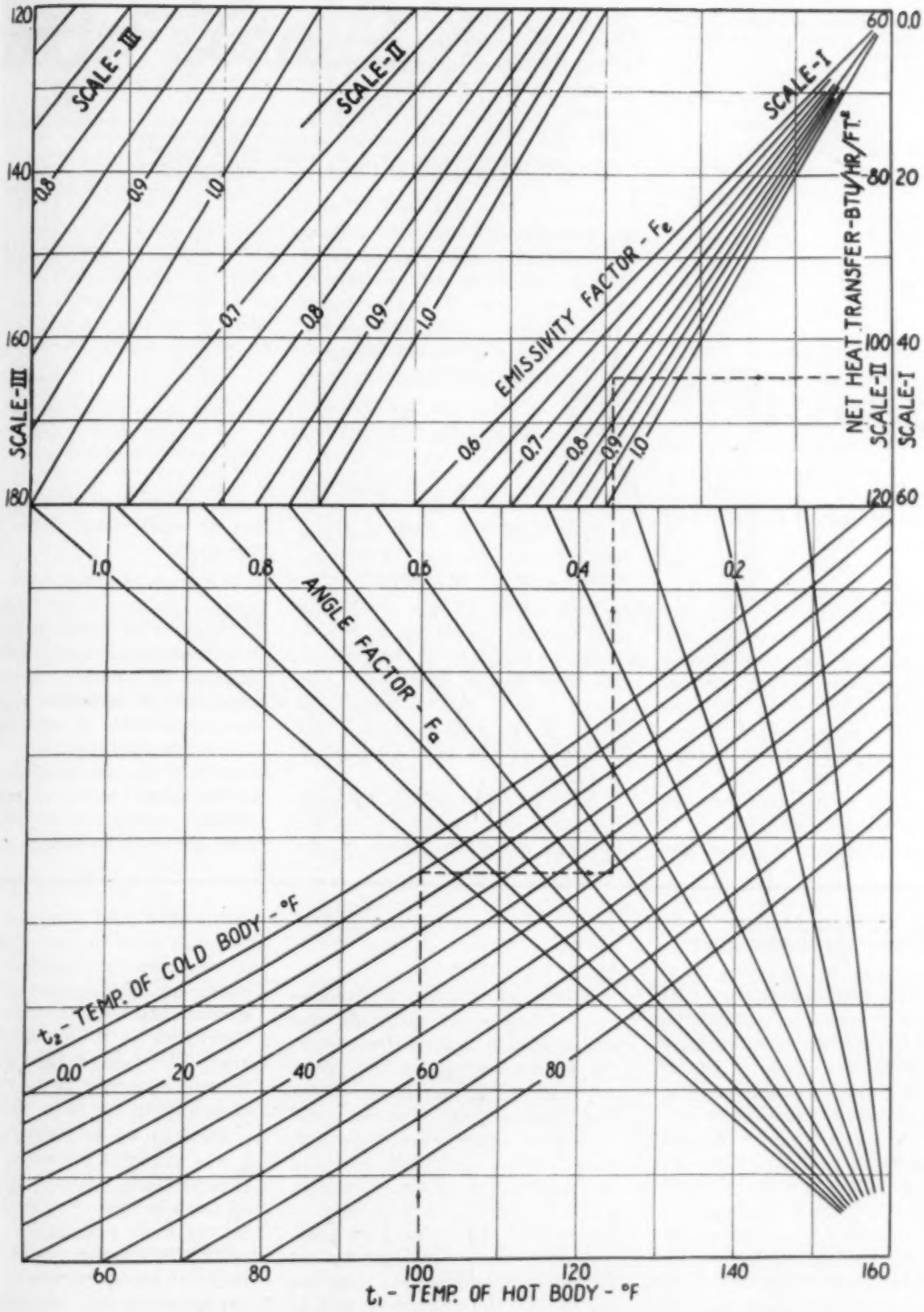


Fig. 2—Chart for graphical solution of the basic radiation equation. The dotted line refers to the example given in the text. Note that in the upper part of the chart, the proper scale I, II, or III on the right or left hand side should be used depending upon which of the diagonal emissivity factor lines in the top section is used

$$F_s = \left(\frac{\cos \phi_s'}{\pi}\right)(\cos \phi_r')\frac{A_r}{r^2}$$

$$= K_s K_r \frac{A_r}{r^2} \quad \ldots\ldots\ldots\ldots\ldots\ldots [4]$$

Equation 4 gives the exact solution for some one position of the line r. When surfaces A_s and A_r are small with respect to the separating distance, values of ϕ_s', ϕ_r' and r do not vary greatly with position of r and hence the shape factor calculated by equation 4 has a value practically independent of the position of line r; for such cases

Table 1—Emissivities of typical building materials in temperature range from 0 F to 120 F

Material	e (average)
Clean metal plate (aluminum, copper, brass), average	0.10
Galvanized sheet iron (average condition)	0.25
Aluminum paint	0.50
Glass	0.80
Linoleum	0.85
Stone (average)	0.87
Wood (unpolished)	0.89
Brick	0.90
Paper	0.92
Plaster	0.95
Asbestos board	0.96

this equation affords a simple, practical means of directly evaluating the shape factor. Table 2 gives values of K_s and K_r for given values of ϕ. Note that in using the table, K_s is read opposite ϕ_s; K_r is read opposite ϕ_r.

Having determined F_e and F_s by the methods indicated above, the net radiant transfer in Btu per hr per sq ft can be obtained from Fig. 2 by entering the lower scale at the known hot body temperature, rising to intersect the line for the cold body temperature, moving horizontally to the intersection with the line for the known shape factor, rising to intersect the emissivity factor line, and moving either right or left to the intersection with the scale for net heat transfer.

Let us determine the heat loss by radiation from a small heating panel (plaster surface, operating at 100 F) located in a wall directly opposite an 8 sq ft window (inside surface temperature 0 F) in a room 20 ft wide.

Example Shows Use of Chart

Solution: For this case ϕ_s' and ϕ_r' have equal values approaching 0 deg, but since the areas are small with respect to the room width, the departure from 0 deg can be neglected; then, from Table 2 for $\phi_s' = 0$, $K_s = 0.3182$ and for $\phi_r' = 0$, $K_r = 1.0$. Therefore, from equation 4,

$$F_s = K_s K_r \frac{A_r}{r^2}$$

$$= 0.3182 \times 1 \times \frac{8}{20^2} = 0.00637$$

From Table 1, $e_s = 0.95$ and $e_r = 0.80$. From equation III, $F_e = e_s \times e_r = 0.95 \times 0.80 = 0.76$.

The solution of this problem is shown by the dotted line on Fig. 2. Since the factor F_s on the graph is 100 times that of the problem, the actual rate of energy transfer is 1/100 of the scale reading $= 0.01 \times 44 = 0.44$ Btu per hr per sq ft of panel.

If, in the above example, the heating panel were in the ceiling, the problem would be solved in the same manner, but ϕ_s' and ϕ_r' would not be equal either to zero or to one another.

Table 2—Source and receiver coefficients for use in the shape factor equation

$$F_s = K_s K_r \frac{A_r}{r^2}$$

Angle ϕ between normal to surface and connecting line, r	K_s	K_r
0	0.3182	1.0000
5	0.3170	0.9962
10	0.3133	0.9848
15	0.3071	0.9659
20	0.2990	0.9397
25	0.2883	0.9063
30	0.2755	0.8660
35	0.2606	0.8192
40	0.2438	0.7660
45	0.2250	0.7071
50	0.2046	0.6428
55	0.1824	0.5736
60	0.1590	0.5000
65	0.1345	0.4226
70	0.1089	0.3420
75	0.0822	0.2588
80	0.0552	0.1736
85	0.0277	0.0872
90	0.0000	0.0000

In certain special cases F_s can be obtained by observation rather than by calculation from equation 4. Whenever one surface is completely enclosed by another (as for the conditions of equations I and II), the shape factor is equal to unity if the source is enclosed, or equal to A_r/A_s if the receiver is enclosed; other similar examples (as parallel planes, concentric cylinders, etc.) will be readily apparent.

INSTALLING PROCESSING MACHINERY OR EQUIPMENT

Amendment to direction 2 to War Production Board order L-41 permits any person to install, without WPB approval under L-41, a single piece or a group of related pieces of processing machinery or equipment, or a single piece or a group of related pieces of any other machinery or equipment approved by WPB on a special application form, if the total cost of the entire installation, including the cost of the equipment, does not exceed $25,000 and if the cost of the job, not counting the cost of the equipment, does not exceed $5000. Building alterations necessary to installations may be made, but no new buildings or additions are permitted by the amendments.

Under interpretation 10 of order L-41, if the cost of the installation exceeds the designated cost limits, permission under order L-41 is required even though the equipment has been obtained on form WPB-541 or form WPB-542.

Heretofore, the cost of processing machinery or equipment was not counted in figuring the cost of the job in determining the necessity for WPB permission under the terms of L-41. Certain large industrial plants were permitted, under certain conditions, to install any number of single or related pieces of processing machinery, if the cost of the installation materials used for each machine did not exceed $500. These provisions are changed by the amendment to L-41 and by the establishment of the $25,000 cost limit in direction 2.

Formerly, the provisions of direction 2 applied only to factories or plants having a productive floor area of 10,000 sq ft or more. The amendment permits the installation of machinery or equipment approved by WPB on a special form, within the cost limits, without qualification as to type of business. It also permits relocation of any machinery or equipment in any industrial plant or factory without restriction as to productive floor area. The former $500 cost limit for materials used in the relocation of equipment has been retained.

Direction 15 to CMP regulation 5 was amended to permit persons engaged in a business designated on the schedules of that regulation to use their maintenance, repair, and operating supplies symbol and rating to obtain materials for work permitted by the amended direction 2 to L-41. Where no construction is involved, the amendment to direction 15 to CMP regulation 5 also permits such persons to use their MRO symbol and rating to obtain up to $500 worth of materials to be used in installing equipment obtained on a WPB special application form.

Sam Lewis' Page

In this regular feature, informal comment on heating, piping, and air conditioning problems and their solutions is given — at the request of the editors — by Samuel R. Lewis, consulting engineer, and a member of HPAC's board of consulting and contributing editors

A READER of this page has asked how to compute cast iron elbow equivalents for thermal circulating hot water heating systems. He sends the accompanying drawing, and inquires particularly concerning:

1) A bull head tee *A* in the 4 in. rigid outgoing header from the boiler, as shown in the sketch.

2) A tee *B* where the expansion tank riser leaves the overhead horizontal flow main. Here there is no particular water movement upward into the expansion tank.

3) A tee *S* in the return pipe at the boiler where the downflowing water may branch out two ways into the 4 in. header.

4) The elbow equivalents in section *Y* to *B*?

5) The elbow equivalents in section *R* to *X*?

Our correspondent makes the problem more complicated and elaborate than is necessary, especially if he should consult the table of elbow equivalents published in the ASHVE's Heating, Ventilating, Air Conditioning Guide, which was developed from research. This table is as follows:

Iron Elbow Equivalents

One 90 deg elbow	1.0
One 45 deg elbow	0.7
One 90 deg long sweep elbow	0.5
One open return bend	1.0
a) One tee, with one-quarter of water diverted out of side	16.0
b) One tee, with one-third of water diverted out of side	9.0
c) One tee, with one-half of water diverted out of side	4.0
d) One tee, with all of water diverted out of side	1.8
One open gate valve	0.5
One open globe valve	12.0
One angle radiator valve	2.0
One radiator	3.0
One boiler	3.0

Ordinary elbow equivalents in length of pipe added are as follows:

Pipe Size, In.	*Equivalent Length of Pipe Per Elbow, Ft.*
1½	3.0
2	4.0
2½	4.5
3	5.0
3½	5.5
4	6.0

Answering item 1, the bull head tee:

The two opposing water currents must create turmoil where they meet, and while such tees frequently are installed in the headers from steam boilers, they are not desirable even with steam, and certainly are not desirable with thermally circulating hot water subjected to the relatively low heat-induced head for creating water movement available in such systems. The tee at *A* should be installed as indicated by the dotted lines, so that the two currents may blend with each other rather than oppose each other.

Answering item 2: This tee (*B*) has all of the water flow out of the side, and is like the value *d* in the table, equivalent to 1.8 elbows. I'd say two elbows.

As to item 3, this tee (*S*) has one-half of the water flowing each way and the condition most nearly approaches value *d* in the table. I'd say two elbows.

Answering item 4, the dotted lines indicate the better layout from *Y* to *B*. Commencing at *Y* on the right, the longer circuit, there are two elbows, then a tee of *c* value, and another elbow, or seven elbows. There will be two more before the changed flow main approaches elbow *C*, if allowance for the tee at the connection to the expansion tank in included.

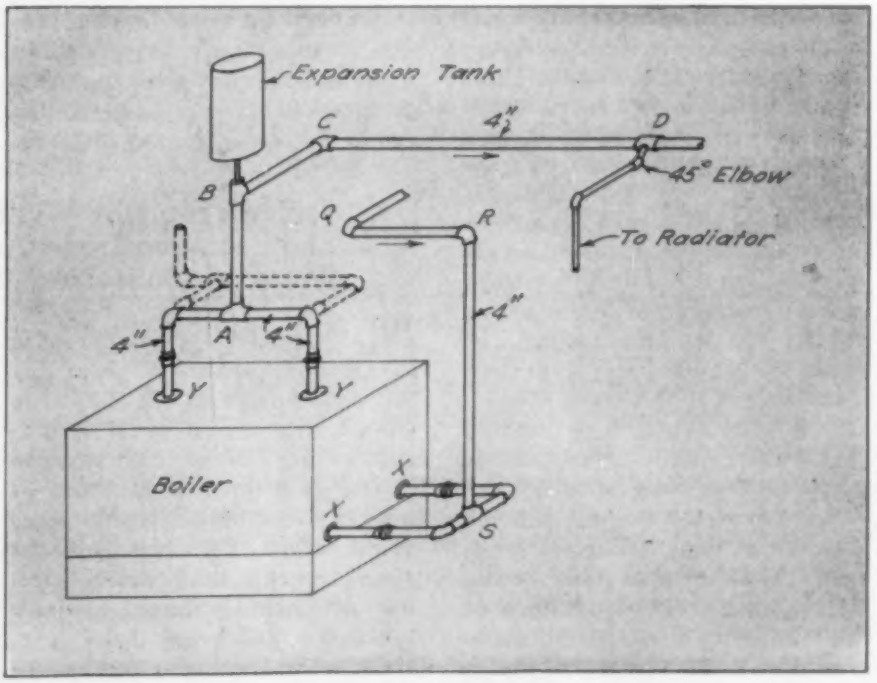
As to item 5, commencing at *R* and running to *X* the circuits are equal and one of them includes the elbow at *R*, the tee at *S* and an elbow near *X*, total four elbow equivalents.

Branches, as shown at *D*, taken off the bottom of an overhead main, are excellent practice. The main may slope downward from *B* so that air will reach the overhead expansion tank.

Ordinarily no one bothers with pitched elbows on hot water piping systems, as the friction added by an extra elbow and nipple is more than compensated for by the improved expansion and convenience in erection due to these fittings.

* * * * *

A reader asks how to compute the elbow equivalents for the fittings shown in this diagram, and is answered here



"OPEN FOR DISCUSSION"

Design and Control of Year 'Round Comfort Air Conditioning Systems . . . Reheating

Discussion by H. M. Hendrickson, Assistant Branch Engineer, York Corp.

I HAVE read with a great deal of interest William Goodman's article on Design and Control of Year 'Round Air Conditioning Systems in the June HPAC, which deals primarily with condenser reheat and the simplified year 'round control obtainable with its use.

Mr. Goodman clearly describes various methods for applying condenser reheat, but leaves so much unsaid that the reader gets the impression that all you need is a cooling system plus condenser reheat in order to have a complete year 'round comfort air conditioning system. This is an entirely erroneous impression, as condenser reheat does not take care of winter heating. Some sort of a heating system is still necessary, as there will be no hot discharge gas available for reheating unless there is a cooling load to keep the refrigeration compressor operating at the same time. Of course, a reverse cycle heating system, with its complication of controls, might be provided by the addition of a winter evaporator cooling outside air or well water if available, but no mention is made in Mr. Goodman's article of reverse cycle operation. Fig. 6 of the article, showing a steam reheating coil, is the only diagram illustrating a year 'round air conditioning system. The arrangements shown in Figs. 7, 9, and 10 for condenser reheat, and the extremely simple controls described in Mr. Goodman's article, are only suitable for summer cooling. A winter evaporator for reverse cycle operation, or a separate heating system, must be added in order to have year 'round air conditioning.

On comfort applications, condenser reheat is normally used for humidity control only (not as a main source of heating) during periods of light sensible load when the moisture load is relatively high. Such conditions in comfort work occur in localities like the Hawaiian Islands and the South Pacific, where the dry bulb temperature is comparatively low with a correspondingly high wet bulb during the greater part of the year. In such regions, little if any heating is required, and dehumidification is the main problem. Such conditions are ideal for the application of condenser reheat, but they rarely occur in the continental United States for a long enough period to justify the added expense of condenser reheat.

An engineering analysis of the principles and limitations of condenser reheat is given in the article by S. F. Nicoll in the ASHVE Journal Section of the December 1940 issue of HPAC, pp. 736-741, to which reference is made in Mr. Goodman's article. Mr. Nicoll points out that condenser reheat finds its application on high moisture loads, particularly for industrial applications, where the humidity must be maintained within very definite limits. By examples worked out mathematically, he shows that the bypass system does a fair job of controlling humidity at light load conditions (probably close enough for ordinary comfort work), but when coupled with condenser reheat, controls humidity exactly and with the lowest overall refrigeration and reheat requirements. Reference to Fig. 1 in Mr. Nicoll's paper, together with his explanation of load ratio lines, will add considerably to Mr. Goodman's discussion of his Figs. 11 and 12.

Condenser reheat, like the bypass system, is a patented arrangement. Its use always results in practically continuous operation of the refrigeration compressor. Its application is limited to compact equipment arrangements with short piping connections between compressor, reheat coil, evaporator and condenser, and with proper allowance for drainage and return of refrigerant condensed in the reheat coil. For year 'round air conditioning, some other source of winter heating—such as steam, hot water, gas, electricity, or reverse cycle—is always required, even if condenser reheat is used. Therefore, condenser reheat does not find much practical application in the ordinary comfort cooling job, unless the humidity requirements are unusually exacting, or the conditions are such that some sort of reheat is always necessary even at maximum load conditions because of an abnormal latent heat load. The refrigerating and air conditioning engineer looks upon condenser reheat as a tool for dehumidifying and for close humidity control, particularly for exacting industrial applications, rather than as a means of heating for year 'round air conditioning.

Author's reply by William Goodman, Consulting Engineer, The Trane Co., and Member of HPAC's Board of Consulting and Contributing Editors

MR. HENDRICKSON'S statement in his second paragraph about my leaving so much unsaid illustrates one disadvantage of serial publication of an article. In writing a series, each installment must of necessity deal with only one phase of the subject. Of course, I did not mean to imply that condenser reheating was to be used for winter heating. I had hoped that the installment in the June issue, as well as the preceding ones, had made plain the fact that I was discussing only the summer phase of temperature and humidity control. The later installments of the article show complete year 'round systems in which steam or hot water coils are used for winter heating. Evidently the articles already published should have carried some statement to this effect.

In any event, would the simple

arrangements shown in Figs. 7, 9, and 10 be any less simple if heating means were added? The same thermostats—without any changeover devices for winter-summer operation—can of course be used throughout the year.

It is difficult to see how during periods of light sensible load, such as occur, for instance, in the intermediate seasons of spring and fall, reheating can be avoided if desirable conditions are to be maintained. In such cases the sensible ratio line can swing to a vertical position. In the past, most air conditioning installations have been made in places such as theaters and restaurants where air conditioning during intermediate seasons was not given much consideration. If people were not altogether comfortable, they were in the conditioned space for only a few hours a week at the most. Even under these contions, however, complaints are still heard today.

In the postwar era, many of the large air conditioning installations are likely to be made in buildings in which people live and work for many hours of the day. Furthermore, numbers of such buildings are likely to be situated in the downtown sections of large cities where noise and dirt are prevalent. In such cases one of the advantages of air conditioning is that it permits windows to be kept closed, shutting out the dirt and noise of the streets. In buildings of this type, the humidity load will be approximately constant through the spring, summer, and fall while the sensible load varies widely. Under these conditions, reheating appears to be necessary if comfortable humidity and temperature conditions are to be maintained during periods of light sensible load.

I do not want to appear in the position of advocating reheating exclusively. There are buildings where the operating cost or the first cost of reheating might make its use impractical. For such cases the primary air system alone, or in combination with reheating for light sensible loads, may prove useful. The primary system, described in later installments of my article, is more limited in its application than reheating. Reheating was discussed extensively in the early installments because it is the simplest and most flexible method of obtaining control of conditions within a building.

In examining the arithmetical example in Mr. Nicoll's paper, to which Mr. Hendrickson refers, I do not find that Mr. Nicoll shows that the bypass "does a fair job of controlling humidity at light load conditions," as Mr. Hendrickson states. Mr. Nicoll shows clearly that under light sensible loads he is *not* able to maintain with the bypass alone the conditions which he sets in his example—conditions which he is able to maintain with reheat. To state that the resulting conditions are "probably close enough for comfort work" is to beg the question. Presumably in a real building the sensible heat load could fall below the minimum set by Mr. Nicoll in his illustrative example. With a bypass, as the sensible load decreases, the amount of chilled and dehumidified air must continually be decreased unless the room dry bulb temperature is to be allowed to fall. With the supply of dehumidified air decreased by means of a bypass, the relative humidity will rise as the sensible load falls. The point of my article is that year 'round air conditioning systems should be so designed that they can maintain desirable conditions for all sensible heat loads.

I make no pretensions to any authority in the field of patent law. However, since Mr. Hendrickson has raised the question, I would suggest for those interested in the use of condenser reheat an examination of French patent 562,075 issued to P. de Dianous in 1923, five years before Shipley applied for U. S. patent 1,837,797 covering condenser reheating.

Mist and Dust Collection ... Static Electricity

Comment by H. C. Murphy, Vice-president of the American Air Filter Co.; Member of HPAC's Board of Consulting and Contributing Editors

I HAVE been much interested in the article on Mist and Dust Collection, by C. E. Lapple, published in the July HPAC. The author shows excellent command of his subject and I feel that these discussions will be of benefit.

In his comments on static electricity, Mr. Lapple considers the explosion hazard set up by friction between dissimilar substances. A research bulletin issued by the Underwriters' Laboratories, Inc., in April 1939 has the following, which might be of interest in this connection:

"From these tests it appears that air practically free from solid or liquid particles is not perceptibly electrified under conditions ordinarily obtaining in blower systems. If dust or lint is present in appreciable amounts, static charges of a magnitude capable of causing ignition by spark discharges may be generated. Voltages actually observed ranged from 100 to over 10,000. It was observed that within certain limits the static charge increased in magnitude with increase in the concentration of dust or lint, fineness of particles, velocity of air stream, and decrease in humidity."

We believe that air cleaners are a worthwhile safety precaution in this connection.

When engineers gather to hear a paper or talk, there comes a time when the meeting is thrown "open for discussion." Some of the audience may offer comment amplifying the speaker's remarks, others will disagree with points the speaker may have made, and some will want to ask questions for the speaker to answer . . . We follow this custom here in these columns, which provide an opportunity for readers to comment and dicuss articles published in HPAC and other topics of interest to heating, piping, and air conditioning engineers and contractors. Your views will be welcomed; address The Editor, Heating, Piping & Air Conditioning, Six North Michigan Avenue, Chicago 2, Illinois.

Names for Things (Nonfreeze Coils)

Comment by W. A. Evans, District Manager, Aerofin Corp.

THERE IS PLENTY of humor in Sam Lewis' comments about Names for Things in the February HPAC, but those terms he jests about seem to find their way around all right. "Coils, traps, fins, gills, gravity, thermal, etc.," seem to be interpreted correctly in their right places. It is not likely that many disappointments are experienced because of misunderstanding of these terms. Not so in some uses of the term "nonfreeze" in connection with blast heating coils.

There appears to be a confused understanding of what a nonfreeze coil actually is and of the actual differences in construction and operation between plain blast coils, nonstratifying design and nonfreeze.

Plain blast coils use single tubes between separate supply and return headers. When steam is full on it is likely to reach all points throughout the face of the coil and give uniform temperature rise. At those times there is little likelihood of freezing if condensate is constantly and fully drained and the coil is well vented at all times. But throttle the steam and cold areas will appear at the most distant points from the steam supply, allowing cold strata of air to pass—with every likelihood of freezing the condensate.

Nonstratifying coils have an external appearance identical with plain blast coils. Their steam and return connections are usually at opposite ends of the unit. They are subject to the same limitations as the plain coil when operating in subfreezing temperatures. Both must be protected against freezing by low limit thermostats set to throw on full steam or cut out the fan when the steam supply falls off or when the air off the coil drops to 35 F. Temperature of the air off the coil must never be allowed to go lower than 35 F. These nonstratifying coils do a better job of distributing steam throughout the entire face of the coil and prevent stratification into hot and cold streams, but they are unsafe in air temperatures below 35 F, in the experience of users.

Nonfreeze coils are both nonstratifying and nonfreeze. The return connection is at the same end as the steam supply and inner distributing tubes are used. Condensate cannot exist even in small quantities except in contact with steam heated surface; that prevents freezing. The flow of condensate is counter to the flow of steam; however the steam may be throttled, there is always sufficient steam in contact with the condensate surface to prevent freezing.

Author's reply by Samuel R. Lewis, Consulting Engineer. Member of HPAC's Board of Consulting and Contributing Editors

CONCERNING MY discussion of Names for Things, there have been some interesting return throws, mostly gentle but some of which required dodging.

W. A. Evans comments above on my omission of the term "nonfreeze convectors." However, since he is a man he is not consistent and persists in his innocence to speak of the gilled convectors used for transferring heat by condensing steam, as *coils*. They are straight tubes, sometimes with little offsets to permit some springing, and I claim they are *not* coils at all.

Mr. Evans' contribution brings out a fact which I had not observed before with adequate detail—that there is so much difference in interior construction between true nonfreeze convectors and nonstratifying convectors. It appears that there are three types of gilled convector adapted to use with fan systems, as follows:

The convector with plain straight or slightly curved tubes between a supply header at one end and a condensate header at the other end.

Nonstratifying convector. This may be identical in exterior appearance with the plain convector, but the supply header delivers steam through distributing tubes within and concentric with the gilled tubes and the condensate in the gilled tubes drains to the header at the tube ends opposite the supply header. It is conceivable that when this convector is placed in an air stream cold enough to freeze, the steam may all be condensed before the condensate header is reached, thus permitting the condensate to freeze.

Nonfreeze convector. Again a superficial inspection of the exterior of the device may not disclose the secret of its ability to withstand frost. The supply and condensate headers are, however, combined at one end of each gilled tube. The tube, being of metal, will not freeze. Only the water can freeze. Water cannot be present unless it is being transported in the state of steam. If steam is present the inner and outer tubes will be warm for as great a distance away from the header as the steam pressure and heat removal conditions permit, and the condensate will be removed before it can freeze.

The Minneapolis-Honeywell Regulator Co. owns a copyright on the term "aquastat," another reader informs me. But the term is so apt that its owner has not, as far as I know, sued any one for infringement. I think it is the inspiration of names such as these that causes my breaking out periodically with a desire for better general nomenclature.

Referring to another comment, "axial flow horizontal discharge unit heater" probably is better than "automobile engine cooling type unit heater." "Axial flow down discharge unit heater" seems to cover the "projection" type. "Radial flow unit heater" might describe satisfactorily the industrial type having several centrifugal fans on one shaft. This heater could be described as horizontal discharge (as when overhead) or vertical upward discharge or vertical downward discharge.

* * *

Arthur Stout has been appointed publicity director of the National Mineral Wool Association, 1270 Sixth Ave., New York 20, N. Y., and will work actively on the association's program to aid the 1944-45 fuel conservation drive.

Heating, Piping & Air Conditioning

AMERICAN SOCIETY of HEATING and VENTILATING ENGINEERS

Journal Section

Looking Forward

By S. H. Downs, President, ASHVE

IN THESE momentous and history making days there is still a lot to be done to speed the winning of the war and as the fall season begins Society members have a responsibility in bringing further public enlightenment on fuel conservation. They have a challenge in several other fields as well. Several that come to mind immediately are: (1) Making the chapter organizations a dynamic influence in their communities, (2) expanding and financing a research program that will contribute to public welfare, (3) extending and increasing the scope and influence of the Society through increased membership.

Steps must be taken in industry, in homes and in every place, where fuel is used for heat and power, to save this vital commodity and this offers both a challenge and a responsibility to all in the heating profession and industry. The chapters can assume leadership in their communities and point out the waste that accompanies overheating and uncontrolled temperatures.

In reports made to the Society at its Semi-Annual Meeting, it was evident that fuel users have only a limited conception of the many factors involved in the proper operation and maintenance of their systems. It was shocking to learn that a survey indicated 20 per cent of heating plants in "good condition," 20 per cent in "fair condition" and 60 per cent in "poor condition." If eight or nine of every 10 systems need attention, the opportunity is offered for a successful war on fuel waste before next winter.

At this time the local chapter officers are planning programs for the fall and winter season. This activity is evidenced by the calls received by the Chapter Relations Committee for program suggestions and speakers.

During October it will be the pleasant duty of the officers to present charters to three newly organized chapters in Denver, Memphis and Columbus, Ohio. In each locality there has been a substantial increase in membership so that the Society has gained greater strength in these areas.

The 50th year will soon come to a close so we look forward to the 51st Annual Meeting at Boston during the fourth week of January 1945, when Massachusetts Chapter will act as host to the Society.

The Use of Glycol Vapors for Air Sterilization and the Control of Air Borne Infection†

By B. H. Jennings,* Edward Bigg, M.D.,** F. C. W. Olson††,
Evanston, Ill.

THE TOLL of air borne disease presents perhaps the largest tax on one's comfort and productive effort. Respiratory infections are responsible for an annual loss of more than 100,000,000 man days in American industry. Control and reduction of this waste represents a real challenge to engineers and physicians.

The concept of the spread of disease by the aerial route is not new, but it was not until recently that cumulative evidence has produced substantiation of this theory. Much of the credit in this field must go to the fundamental work of Wells,[1] who showed that droplets delivered from the respiratory tract by coughing, sneezing or talking, evaporate rapidly and become *droplet nuclei* which float in the air for long periods, finally settling to the floor in the room. Here they adhere to dust particles and can again be set into motion by air currents. The development of simple, yet reliable devices for quantitative collection of bacteria from the atmosphere has also simplified and furthered bacteriologic studies on this problem.

A number of workers have reported evidence, both clinical and bacteriologic, on the spread by air transmission of specific infections[2] such as hemolytic streptococcal infections, measles and mumps. Laboratory experiments have also proved that infection of susceptible animals occurs when they are exposed to atmospheres containing pathogenic organisms.[3]

Air borne disease may be controlled by three methods: (1) Production of immunity in exposed individuals by vaccines, serums, drugs, etc., (2) prevention of introduction and dispersal of disease-bearing organisms into the atmosphere, (3) reduction in the number of organisms in the air of enclosed spaces.

A discussion of the first two points is not within the scope of this presentation, but it may be said that to date these measures leave much to be desired, although great strides in their development have been and are being made.

It is obvious that frequent air exchanges in a space will, by dilution, reduce the total bacterial contamination of the space. This is the commonest means of bacterial control in use. However, in addition to the fact that the method is not positive there are limits to the number of air changes possible. Filtering and electric precipitation of dust and organisms in air also reduce contamination but fail to achieve complete control.

The bactericidal activity of ultraviolet irradiation is well established. If micro-organisms are exposed to the direct action of the rays, death is quickly produced. There are, however, certain objections to the widespread use of this means of air sterilization. Since exposure of the skin and eyes to ultraviolet rays may produce undesirable effects, individuals in the treated space must be shielded from the direct rays. This necessitates irradiation of only those zones in which the rays can be directed without interference with the persons in the room. Although the bacterial content of such zones is lessened, there is a remaining area in which direct droplet infection may occur and in which the bacterial counts are high. The total effect is nevertheless a reduction in air contamination since air currents ultimately expose all the air to the irradiated area.

The use of ultraviolet lamps is further restricted by cost of installation and operation. The elements must be changed at intervals because after a certain period the wavelengths of the emitted rays are no longer bactericidal. Greatest germicidal activity is exhibited when dried organisms are exposed to the rays; the effect on moist droplets is minimal. There have been, however, several reports on clinical trials of this method which suggest that under certain conditions it may be effective.[4]

Sterilization of air by germicidal mists was attempted many years ago by Lister, who sprayed operating rooms, with phenol solutions, but further development of this method of bacterial control awaited the development of chemicals which fulfilled certain criteria. These include: (1) Non-toxicity, (2) high bactericidal activity in low concentrations, (3) imperceptibility to the sight, taste, and smell of the individual exposed, (4) availability in large quantities at low cost, (5) ease of introduction into and maintenance of concentration in the treated space.

The first practical approach to this problem was made by Douglas, Hill and Smith in 1928.[5] This paper dealt with the use of atomized solutions of *NaOCl*. They were able to produce marked reduction in bacterial counts of air-suspended B. coli using a concentration of 0.5 mgm *NaOCl* per liter of air.

Ten years following this report, Masterman[6] extended these earlier observations and reported sterilization of air by 0.02 mgm of 1 per cent *NaOCl* in 1 liter of air. He

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†The work described in this paper was done under contract recommended by the Committee on Medical Research between the Office of Scientific Research and Development, and Northwestern University.

*Professor of Mechanical Engineering, the Technological Institute, Northwestern University. Member of ASHVE.

**Instructor, Department of Medicine, Northwestern University Medical School.

††Research Associate, Northwestern University.

[1]Superior numerals refer to Bibliography.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Grand Rapids, Mich., June 1944.

stated that the germicidal effect was produced by molecular *HOCl* liberated from the sprayed droplets of *NaOCl*. A concurrent report by Trillat[7] appeared in 1938. After experimentation with the atomization of the common germicidal agents he concluded that only resorcinol and sodium hydrochlorite were satisfactory. Trillat also propounded the theory, which is now questioned, that these substances exert their effect because of their physical state, *i.e.*, as aerosols. In brief, he believed that each droplet of sprayed bactericidal compound retained the concentration of the parent solution and because of its small size (1 to 2 microns) remained suspended in the atmosphere, where it ultimately collided with droplets containing bacteria, thus acting in the same concentration as the original solution.

Continuing observations were carried farther by certain English workers. Pulvertaft and Walker[8] advised solutions of resorcinol, glycerol, and water. Twort[9] and his associates after a comprehensive study of the subject developed a solution containing hexyl resorcinol, loral sulfate and alkaline propylene glycol which they found highly effective. Both groups of investigators based their experiments on Trillat's aerosol theory and for this reason attempted to find means for the prevention of evaporation of droplets. This explains the first use of glycerol and propylene glycol which because of their low vapor tensions and hygroscopicity maintained droplets in air for longer periods.

Properties of the Glycols

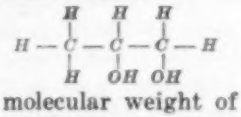
Recently the glycol group of compounds in vapor form have been found to be highly effective agents for air sterilization.[10] Although all glycols possess killing power in varying degrees, propylene glycol (*PG*) and triethylene glycol (*TEG*) fulfill most completely the criteria noted above for widespread use. These materials had been used industrially as solvents and dehumidifying agents for many years, but it was not known that they presented any germicidal value. In fact, they exhibit relatively low bactericidal activity in test tube experiments. Cultures of micro-organisms grow readily in broth containing 15 per cent propylene glycol and death does not occur until the glycol concentration of the media reaches 75 per cent or over. However, when dispersed in the air, minute quantities in vapor form exhibit a dramatic killing action. It might be mentioned at this time that this latter observation tends to disprove Trillat's aerosol theory. Since death is produced almost instantaneously in air suspended organisms, this phenomenon cannot be explained by collision of bacterial and glycol droplets, but may be computed on the basis of interaction between molecular glycol and bacterial droplets.

Toxicity Studies

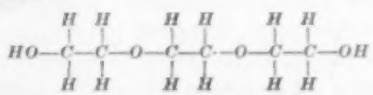
PG and *TEG* have the lowest toxicities of any compounds suggested for air sterilization. Comprehensive studies on ingestion and parenteral administration show that the toxicity levels are somewhat below that of ethyl alcohol.[11] Observations on exposures of rats, guinea pigs, and monkeys to prolonged periods of inhalation (approximately two years) have shown no local or systemic evidences of toxicity.

Physical Characteristics of PG and TEG

Propylene glycol, $C_3H_6(OH)_2$ has a structural formula



with a molecular weight of 76.06, boils at 368 F and has a vapor pressure of 0.016 mm of mercury at 68 F. Triethylene glycol, $C_6H_{12}O_2(OH)_2$ has a structural formula



with a molecular weight of 150.17, boils at 548 F and has a vapor pressure of 0.00083 mm of mercury at 68 F. It can thus be seen that these are *high-boiling-temperature dihydroxy-alcohols*. They are freely miscible with water and highly hygroscopic. They are essentially odorless, both in liquid and vapor form. In very high concentrations the vapors give an impression of sweetness. Because of the nontoxic character of *PG*, which was first established, much of the early experimental work involved the use of this material. However, when *TEG* was also shown to be nontoxic,[11] it almost completely displaced *PG* for bactericidal use as only about one fiftieth as much *TEG* is required for the same bactericidal effect.

Fire Hazard

The inflammability characteristics of *PG* and *TEG* have been studied and reported elsewhere.[12] It has been found that:

1. The vapor-phase concentrations required for air sterilization were completely free of any fire hazard.
2. The presence of water in combination with glycol greatly reduced its combustibility. This was particularly significant, since samples of condensed material from surfaces as cold as —20 F never contained more than 20 per cent glycol. It is impossible to ignite such solutions even with prolonged application of heat.
3. The introduction of small quantities of water with glycol renders large storage tanks relatively inactive as far as inflammability is concerned.

Small Chamber Experiments[10]

These experiments were carried out in small glass chambers of two cu ft capacity. Bacterial cultures were sprayed into the space by glass atomizers and measured quantities of air were removed for quantitative determination of bacterial content. Since there is a constant spontaneous reduction in recoverable organisms from the atmosphere after spraying, a control chamber was used in each experiment. Agitation of the air to insure mixing was accomplished by the use of small revolving fans. Organisms tested and found to be susceptible to the action of glycol vapors were pneumococci, hemolytic and non-hemolytic streptococci, staphylococci, Escherichia coli, H. influenzae, B. pertussis, and Streptococcus viridans. Careful controls were carried out to show that actual death for the test organisms occurred.

To demonstrate the virucidal activity of *PG* and *TEG*[13] mice were placed in the test chambers and a mouse-adapted strain of influenza virus (P.R.8) was then atomized into the space. Complete protection of the exposed mice occurred in the glycol treated chambers in contrast to almost 100 per cent death of mice in the untreated chamber.

Conclusions drawn from the above experiments were: (1) In concentration of 0.2 mgm *PG* per liter of air and 0.005 mgm *TEG*

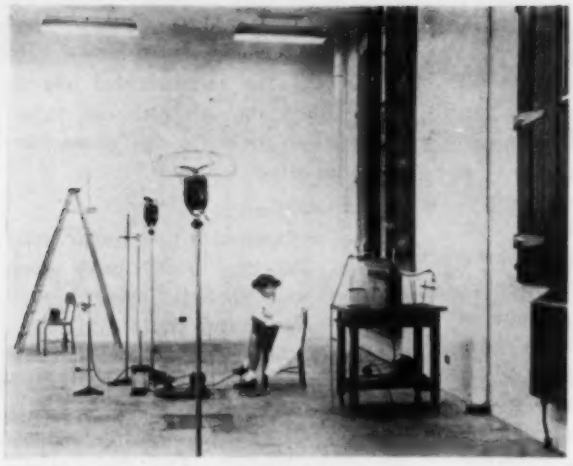


Fig. 1—Experimental test room

per liter (0.0013 to 0.0021 grains per cu ft)‡ of air immediate death of air-borne micro-organisms is effected, (2) relative humidities of 25 to 60 per cent are necessary for optimum activity, and (3) moist droplets are more susceptive to glycol actions than dried organisms.

Tests in Large Unoccupied Space

Studies preliminary to large scale application were conducted in an air conditioned test room of the Technological Institute of Northwestern University shown in Fig. 1. This room is 38 ft long, 17 ft wide and 16 ft high, with an approximate volume of 10,000 cu ft. It is insulated, extremely tight in construction and with special sealing reduces air infiltration and exfiltration to a low minimum. Large refrigeration type doors were used for entry and over each of the windows. This enabled us to carry out our experiments without the factor of large quantities of uncontrolled air. Tests were made to: (1) Demonstrate the lethal action of glycol vapors in this large space, (2) test the effectiveness of vapors produced by different methods, and (3) check the distribution of glycol vapors in a room of significant capacity.

The results of these experiments have been previously reported,[13] but it would be of interest at this time to review the findings.

Test organisms used were *Staphylococcus albus* and a guinea pig strain of hemolytic streptococcus (streptococcus C). Bacterial suspensions were made in filtered saliva. Saliva was used as the diluent since it was noted that organisms suspended in this material remain in the air for long periods of time. This simulated field conditions as closely as possible. The suspensions were sprayed by means of an atomizer which produced an extremely fine mist.

Bacterial samples were obtained by a Moulton air bacterial sampler[14] and reported as numbers of colonies per cubic foot of air.

A test was run as follows: The room was first thoroughly aired by opening the windows and doors and turning on two large propeller type circulating fans placed in the room. It was then made air tight and 10 cu cm of the bacterial suspension were sprayed directly under the two circulating fans, at opposite sides of the room. Bacterial air samples were taken immediately after the cessation of the spray and at intervals of 15 min for a total period of 2 hours. The doors and windows were then reopened, fans speeded up and the room allowed to air for one hour. The space was again sealed, glycol vapor introduced, and 10 cu cm of the same bacterial suspension used for the control test were sprayed under the fans. Samples of air with the same time intervals as before were taken to determine bacterial content. Glycol concentrations existing in the air were also measured. Results of the tests corroborated the small chamber experiments and the same striking bactericidal activity of *PG* and *TEG* vapors was demonstrated. The previous observations on relative humidity requirements were also substantiated. At levels of 80 per cent or over there was practically no lethal effect. Fig. 2 is a photographic reproduction of a typical test.

Carefully conducted observations on the behavior of the vapors were made and showed that if air exchanges were kept at a minimum, the glycol concentration decreased at a slow rate. Data on propylene glycol show that it took more than 4 hours in the test room for the glycol concentration to reduce from 0.22 mg per liter of air to half that amount. There was evidence of a slight amount of absorption on room surfaces in addition to the larger amount removed under the small but existent air exchange. Only when excessive concentrations of the vapor were produced was the presence of the material perceptible to the laboratory workers. Observations were made to determine whether any evidence of stratification of glycol vapor at different levels in the quiescent room existed. Tests running over a 30-hour period showed no significant variation in glycol concentration at different levels in the test room.

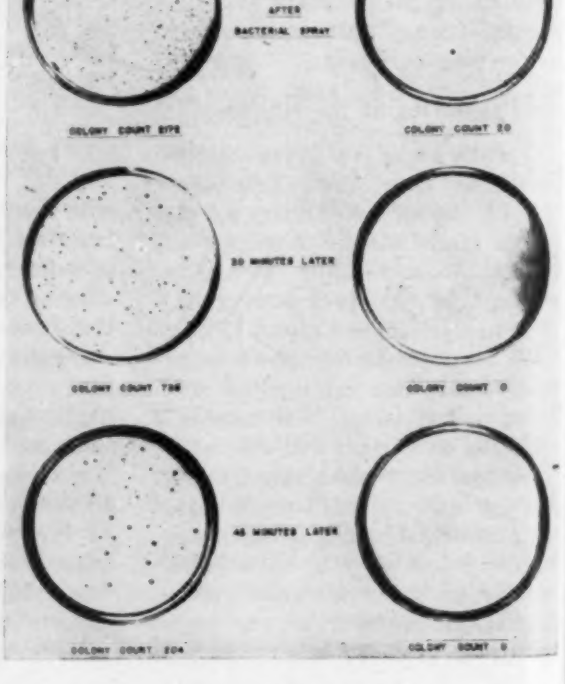


Fig. 2.—Bacterial plates from test room (Staphylococcus albus)

‡For further clarification this may be expressed as an average bactericidal concentration of 0.67 parts per million by volume or 3.42 parts per million by weight.

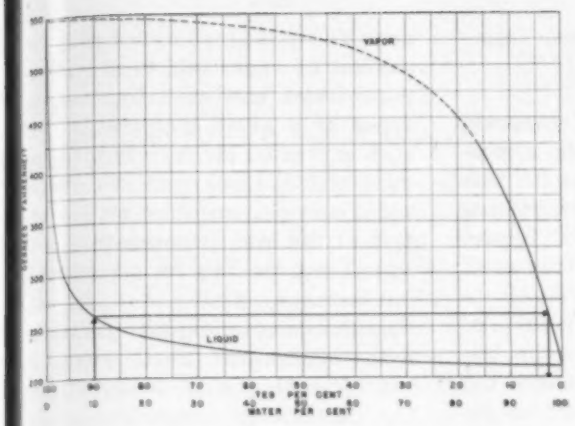


Fig. 3 — Temperature composition diagram of triethylene glycol

Means for Generation of Vapor

Before application of the germicidal effect of these vapors could be made, it was necessary to develop practical means for their generation. Such apparatus had to be of sufficient size and capacity to treat large spaces and incorporate control devices so that adequate sterilizing concentrations could be built up and maintained.

The problem of putting glycol vapor into the air can be solved in several different ways, one of the most satisfactory of which was found to be by vaporization from an aqueous solution. *TEG* is extremely hygroscopic and miscible with water in all proportions. As is true of miscible binary mixtures, the temperature at which boiling takes place varies with the concentration of the particular mixture. At atmospheric pressure, solutions high in water will vaporize at a temperature above, but close to, 212 F. Solutions of *TEG* rich in glycol and low in water will vaporize closer to the boiling temperature of pure glycol, 548 F. When a glycol-water solution is heated, water, being the more volatile component, vaporizes more readily; however, both water vapor and glycol vapor are delivered from the boiling mixture. Reference to the composition diagram of Fig. 3 shows, for example, that an aqueous solution of *TEG* with a boiling point of 262 F delivers vapor containing about 3 per cent glycol and 97 per cent water. This water-vapor delivery does not constitute a disadvantage but is desired since, as has been noted above, relative humidities ranging between 25 and 60 per cent are required for optimum bactericidal action in the space which is to be treated with glycol vapor.

As boiling of such a mixture continues and water leaves at a more rapid rate, the remaining liquid becomes richer in glycol and its boiling temperature rises. If the heat supply is continued, ultimately pure glycol will remain. From a given vaporizer vessel, in an operating range of 240-290 F, the amount of glycol removed is so small that it requires infrequent replacement while water must be added almost continuously.

The concept of maintaining a constant glycol concentration became the basis of the control system for the vaporizer.§ As the device operates, and its temperature rises to a given point, a thermostat set at that temperature opens a solenoid valve which in turn feeds water into the vaporizer in amount sufficient to lower the vaporizer temperature to the point at which the valve is closed. Thus by action of a simple thermostat controlling a water feed valve, the boiling temperature can be kept in a narrow range and the relative rate of water and glycol delivered to the air accurately controlled. Varying the rate of heat input controls the rate at which water and glycol are delivered to the air, but does not vary the ratio of glycol to water vapor. Any desired ratio can easily be obtained by merely adjusting the temperature at which the thermostat is set to operate. A high thermostat setting gives a larger percentage of glycol vapor, a lower thermostat setting, a relatively larger percentage of water vapor. The functions are thus performed in this manner: *change thermostat setting to vary relative water-glycol vapor delivery, change heat input to vary quantity delivered.*

§Patent application made and assigned to the Secretary of War.

Fig. 4—Saturation density of triethylene glycol vapors

The simplicity of this control and its effectiveness have been well demonstrated in field tests in which the authors have been able to build up or reduce the glycol concentration in the space by application of the principles outlined above. With constant wattage input at the electric heater it should be observed that an increase in thermostat setting will permit the heat to work to a greater extent on glycol vaporization, and consequently build up the glycol concentration in the space, although under these conditions less water is supplied.

In all authors' field tests a vaporizer was used. The first tests were made using a laboratory model vaporizer, but the later tests were made with a commercially manufactured vaporizer made to authors' specifications. This is pictured in Fig. 5 and consists of a double shelled unit with the space between the shells filled with suitable insulation. The vapor is discharged from a pipe which attaches into the vapor space of the unit just beneath a removable top. This outlet can be piped to lead the vapor into a duct system or if the vapor is delivered in front of a fan it can be distributed by air currents throughout a small room. An indicating thermometer is placed at the lower front of the unit and above this is a 1000 watt immersion heater. To the right of the heater is a temperature control switch which is adjusted to open the solenoid valve. Thus when the temperature rises to a certain point, indicating that the water concentration in the solution is too low, this switch opens

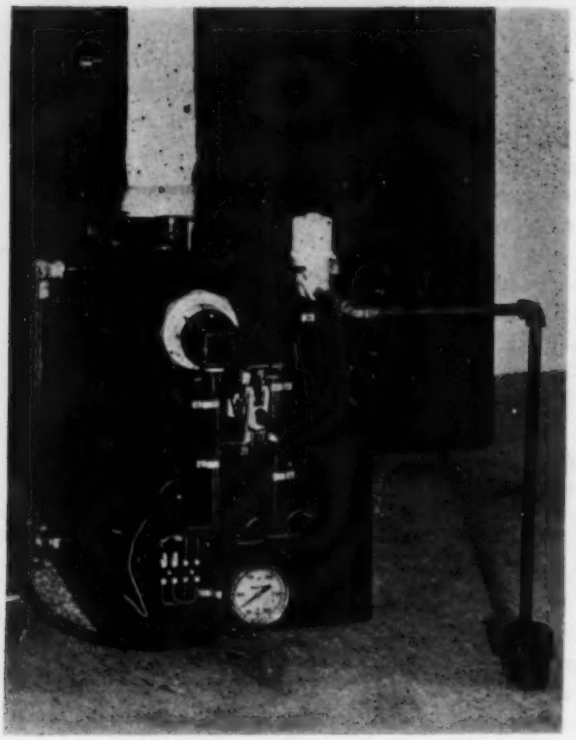


Fig. 5—Glycol vaporizer

the solenoid valve, which remains open until the boiling temperature drops to a low enough point for the temperature control switch to operate and close the solenoid valve. At the left of the heater element is a high-temperature safety cut-off switch to protect the vaporizer from overheating. Near the top in a central location can be seen a high-level liquid float control which trips the electric circuit in the event that the liquid level becomes too high. Below this is a manual reset which has to be used to put the unit back in service when either the float switch or high-temperature cut-off operates. A gage glass to show liquid level is attached.

In addition to investigating the characteristics of a vaporizer,[15] a large amount of work was done concerning the use of an atomizing device or scrubber unit for putting glycol into the air.[16] Under this method an atomizing nozzle forces a mist of the aqueous glycol solution into a moving air stream. Eliminator plates following the nozzle remove the liquid particles, which drain back to the pump reservoir. Water and glycol vaporize into the air stream and are delivered by the attached duct system to the treated space.

There is a certain point at which glycol vapor will be sufficient in amount to saturate completely the atmosphere. Above this point if additional quantities of glycol are present they must exist in suspension in a foglike condition. In the case of the atomizing device the amount of vapor which passes into the air will depend upon the air temperature, and a fog in the treated space would not arise unless the air temperature leaving the atomizer drops appreciably. For any aqueous solution being sprayed into passing air there is a certain relative humidity of water and glycol vapor concentration in equilibrium. The control of such a unit thus hinges about the use of the proper aqueous solution to give the desired glycol-humidity equilibrium in the treated space. Automatic controls for this purpose are being worked on but have not yet been perfected. It will be realized that air with a low glycol vapor content entering can carry only a certain maximum content of glycol vapor as it is discharged. Thus with such a system, using recirculated air and an atomizer, the glycol concentration in the air will rise to the desired maximum which can be reached only by vaporization into the airstream for the particular air temperature at which operation is being carried on.

The development of a sensitive means of detection of glycol in the air which would control the operation of the glycol generator is necessary for complete maintenance of desired room concentrations. Such devices are being studied. Until they are available one must compute the approximate number of air exchanges and introduce vapors at a rate sufficient to compensate for exfiltration. It is believed, however, that the final development of the air scrubber will operate to maintain constant concentrations.

In Fig. 4 are data on air saturated with triethylene glycol. Saturated in this sense means that the amount of vapor held is the maximum which can exist in space as vapor at a given temperature. If such saturated air is cooled, fog or mist would appear as the excess vapor over that required for saturation at the lower temperature condenses out as fog or precipitates on surfaces. Notice that the saturation density rapidly increases with temperature. Fortunately, under desired inside temperatures, the bactericidal concentrations of the glycols required are far below the saturation or fog conditions in air and the glycol cannot normally be detected by the occupants in any way.

Propylene glycol, as before mentioned, is also an effective air sterilizing agent, but it is needed in much greater quantities, namely, between 0.1 to 0.3 milligram per liter (0.044 to 0.13 grain per cu ft) for effective bactericidal action. These concentrations also are far from saturation so that it can be used with little probability of detection by the occupants of a treated space. However, inasmuch as from 40 to 60 times more propylene glycol is required than triethylene, the latter is replacing it in favor.

A few simple mathematical considerations will be helpful in understanding the manner in which glycol concentrations may be built up in a room.

For example, when:

w = pounds weight of glycol vapor in room.
a = number of air exchanges per hour.
b = glycol output of vaporizer in pounds per hour.
t = time in hours.

then

$$\frac{dw}{dt} = b - aw,$$

the solution of which, assuming an initial condition where $w = 0$ when $t = 0$, is

$$w = \frac{b}{a}(1 - e^{-at}).$$

The maximum value occurs, therefore, when

$$w = \frac{b}{a}$$

As long as there is some air exchange taking place in the room, the glycol concentration will increase to a fixed value but cannot increase indefinitely. Also the

greater the number of air exchanges, the sooner the final concentration will be reached.

Tests on Large Occupied Spaces

For this purpose eight dormitories were selected, each housing approximately 80 men. Four of these were used as test spaces and four were used as controls. Each room was approximately 120 ft long, 30 ft wide, and 9 ft high, having a capacity of approximately 33,000 cu ft. A double row of lockers extended the length of the room, effectively dividing it in two. It was, therefore, necessary to use a duct system to distribute glycol and water vapor throughout the space.

Since the distribution of glycol vapors in a duct system presents some new engineering problems, a full scale model was constructed and tested in the Institute laboratory before field installations were made.

Laboratory Tests

The duct used in these tests was of laminated asbestos construction, specially treated on the inside to resist absorption of glycol. It consisted of three 33-ft sections 8½ in. x 14½ in., 8½ in. x 8½ in., 8½ in. x 5½ in., inside dimensions. The transitions from one size to another were of truncated pyramid shape, about 6 in. long. The far end of the duct was closed and the near end connected to the fan unit as shown in Fig. 6. Three turning vanes were placed in the right angled elbow to straighten the air flow into the duct.

Instead of the customary louver and baffle arrangements, special *venturi* openings were used for distributing the air-water-vapor-glycol mixture. These openings were plaster cast fittings having a 2 3/32 in. diameter free opening, faired outward and inward to give a venturi-like cross section. The duct was equipped with 36 such outlets, 18 on each side, spaced at approximately 5-ft intervals. Starting from the elbow, the first outlet was placed 10 ft 4 in. from the inside turn of the elbow on the right side of the duct; the second outlet was placed one foot farther on the left side of the duct.

The arrangement of the fan unit may be observed in Fig. 6. It consisted of three coaxially mounted blower fans directly connected to a ¼ hp motor. The six speeds of the motor produced from 750 to 1700 cfm at free delivery of the fan. The speeds were controlled by a selector switch on the side of the cabinet. The front of the unit is provided with a louver and damper for controlling recirculated air. The back is connected to the fresh air intake and likewise provided with dampers. Both sets of dampers are operated simultaneously by a shaft extending through the top left side of the cabinet. With these controls the fan output can be varied uniformly from 100 per cent outside air to 100 per cent recirculated air. A heating element was also provided in case it became necessary to heat some or all of the incoming air. In these tests the heating element was not used. Provision was also made for inserting air filters just above the heater.

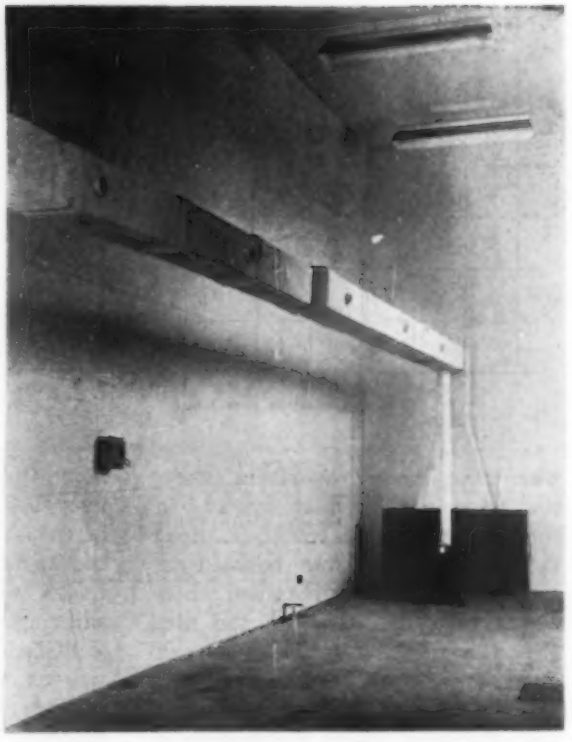


Fig. 6—Assembled duct for glycol distribution (temporarily assembled for photographic purposes)

The first test was on fan output with adapter and elbow. This corresponds roughly to finding free delivery characteristics at the entrance to the duct. A short length of duct (8 in.) was attached to the elbow and anemometer readings taken at various fan speeds. The results are given in Table 1.

Table 1—Fan Output under Free Discharge

Speed No.	Average Air Velocity (fpm)	Output (cfm)
1	1998	1690
2	1369	1160
3	1118	950
4	1000	850
5	865	735
6	830	706

To obtain the relative air velocities through the various outlets, anemometer readings were taken with the back of the anemometer case flush with the opening. A true air speed reading is impossible under such conditions since the outlet is smaller than the case, but an accurate comparison of the relative outputs is readily made by this method. The results for various fan speeds are given in Fig. 7. The numbers in the chart refer to the outlets in progressive order from the fan end. It can be seen that fairly uniform air velocities are obtained. The system was further checked by the use of a smoke bomb. This showed the air to be blown 20 ft horizontally from the outlets before an excessive diffusion occurred. Since the openings were only 5 ft apart, it seems certain a uniform air distribution in the room was produced.

Velometer and draft gage readings were also taken at each outlet and for each fan speed. The results are in substantial agreement with the anemometer readings. The air velocities in the duct immediately after the right angle turn computed from velometer readings at the outlets were found to be 1680 fpm for speed No. 1, 1070 fpm for speed No. 3, and 840 fpm for speed No. 5.

The glycol-water vapor output of the laboratory model vaporizers was determined by weighing the vaporizers with their contents before and after a run of 48 to 72 hours. The vapors escaped into the open air through a ⅜ in. pipe size tapping in the cover. However,

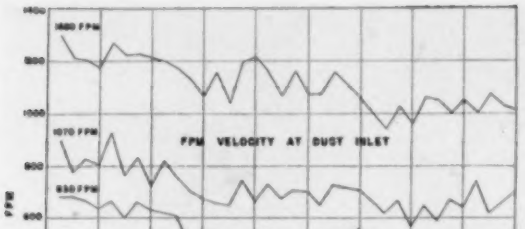


Fig. 7—Anemometer data on outlets

when the same vaporizers were connected to the duct by a 4-ft length of well-insulated ⅜-in. pipe, the glycol output was reduced to about one-tenth that obtained when discharging directly into the air, although the water vapor output was not affected. This reduction was found to be due to the small pipe diameter which resulted in a high ratio of wall area to cross sectional area. Since glycol vapor condensed very rapidly on the walls of the small pipe, despite adequate insulation, a reflux action was produced. When the pipe size was increased to 1¼ in. there was no appreciable loss of glycol output. All of the work indicated the necessity for good insulation and ample pipe size.

Analyses[17] were made of the glycol content of the air coming from each opening and at all fan speeds. These showed that uniform mixing of glycol occurred before reaching the first opening and that there was no perceptible loss of glycol as the air flowed through the duct.

Field Installation

For the field test, the test model was installed in a test dormitory and similar units constructed for three other dormitories. The only modification made was the replacement of the first 33-ft length of duct with one having an 8½ in. x 11½ in. cross section instead of 8½ in. x 14½ in., as in the test model. The appearance of the installation is represented diagrammatically in Fig. 8. For clarity, the double row of lockers in the center of the room and several bunks in the left foreground are not shown. The duct was hung on the center beam and fresh air brought in through an adapter placed in the window frame. A water line was connected to the solenoid valve in the vaporizer.

For adequate comfort it was decided to allow not less than 25 cfm outside air per individual. Since it was estimated that the uncontrolled normal infiltration of air was equivalent to two air changes per hour, the fan speed was set to deliver between 1000 and 1200 cfm of outside air. As far as possible, the doors and windows of the test rooms were kept closed and all air entering the duct was *glycolized*. Ventilation in the control rooms was at the discretion of the occupants; it is believed that the number of air exchanges were greater than in the test quarters.

Since the volume of the room was 33,000 cu ft with the equivalent of four air changes per hour, it was calculated that a glycol delivery of 0.04 lb per hour would be needed to maintain a concentration of approximately 0.004 mgm *TEG* per liter of air. To obtain this output the operating vaporizer temperature was set at 280 F. This setting delivered a concomitant water delivery of 2.5 lb of water per hour.

The dormitories were heated by steam radiators with a single outside-inside thermostat controlling the flow to four rooms. The buildings used were so arranged that each thermostat regulated the temperature of two test and two control spaces. This presented a difficult problem in the conduct of the studies, particularly so as there were no individual cut off valves on the room radiators. Thus the maintenance of adequate glycol concentrations was frequently interfered with by unauthorized opening of windows because of excess temperatures.

It might be noted at this time that the room temperature in the treated space averaged 2.9 F higher than in the control space and ran between 74 F and 76 F during the greater part of the time.

Results

Some operating difficulties were encountered, but these were not serious. It was necessary to add glycol to the vaporizers at three- to four-week intervals and no other servicing was required. Condensation and resultant dripping of liquid glycol from the first duct section occurred at irregular intervals and was most disturbing during periods of extremely cold weather. Tempering of the incoming fresh air would have eliminated this trouble. However, due to the unsatisfactory control of heating system in use this could not be done.

Anemometer readings on air flow and glycol output from various duct openings agreed with the laboratory observations. Glycol distribution

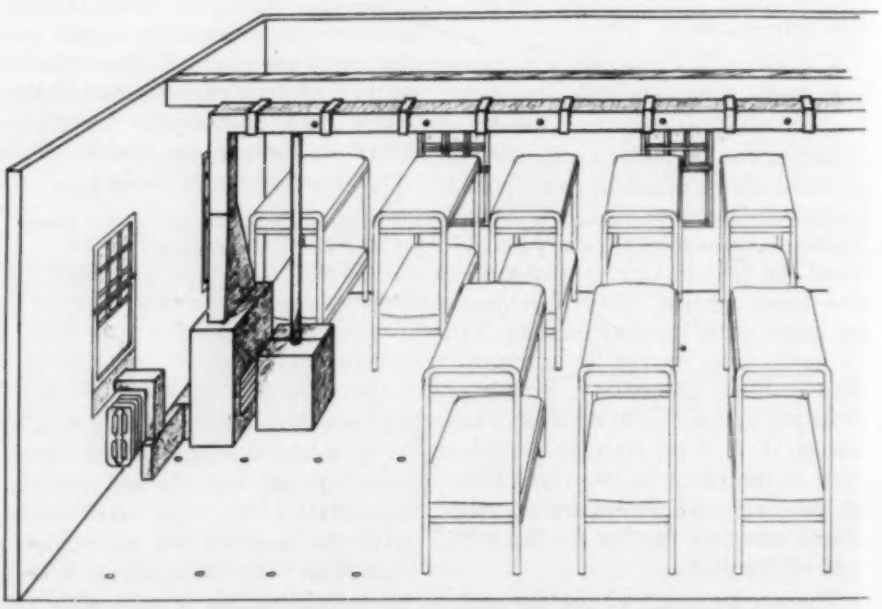


Fig. 8—Field installation for air sterilization in dormitory

in the rooms was uniform. Glycol determinations were made at 2 a.m., 5 a.m. and 8 a.m. The average concentrations were found to be 0.002 to 0.003 mg per liter of air. These values would probably have been higher if duct condensation had not occurred and closure of all windows and doors been rigorously enforced. In a modern air conditioned space with controlled air exchanges, temperature and humidity, little difficulty in obtaining a desired concentration would be expected.

An average relative humidity of 35 per cent was maintained in the test rooms during the test period. This value is in the optimum range previously mentioned.

The occupants of the test dormitories made no complaints regarding the odor of glycol vapors. Some were able to detect its presence in the room immediately when entering from the outside, but quickly became accustomed to it.

A marked reduction in the number of air borne bacteria was obtained in the test rooms. Results for one test period are shown in Fig. 9. There are several reasons why the spectacular reduction shown in Fig. 2 did not occur. As has been mentioned, the glycol concentrations were somewhat lower than desired. A more important reason is that the bacteria normally present in air include many resistant types as spore formers and various encapsulated organisms. In addition, it is believed many air borne bacteria are attached to dust particles, in which case sterilization may be more difficult.

Records were kept of the number of men from each dormitory contracting diseases believed to be air borne. Studies of the hospital records indicated a noticeable reduction of infections in the test dormitories.¶ The results were very encouraging and further carefully controlled observations with larger groups are being planned.

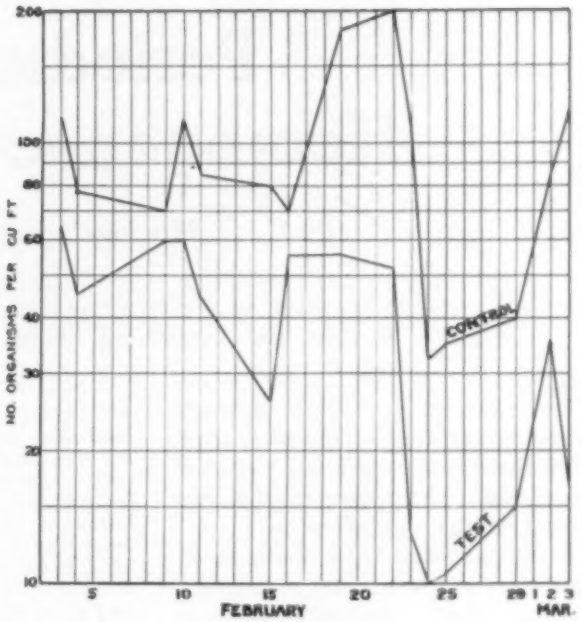


Fig. 9—Bacterial count data—Each point represents an average of twelve readings

Conclusions

1. Bactericidal quantities of glycol vapors can be maintained in large occupied spaces for long periods of time.

2. It has been shown that glycol vapors can be uniformly distributed in large rooms.

3. A means for generating glycol vapors in controlled amounts is described.

4. The occupants of treated rooms experience no discomfort due to glycol vapors.

5. Triethylene glycol is preferred over propylene glycol because much smaller amounts of the former are bactericidal.

6. Optimum bactericidal concentrations required with triethylene glycol range from 0.003 to 0.005 mg per liter of air (0.0013 to 0.0021 grains per cu ft).

7. Relative humidities in the range 25 to 60 per cent are required for optimum bactericidal action.

¶A complete analysis of these data and the medical aspects of these tests is to be published elsewhere.

Acknowledgments

A. F. Hubbard and R. H. Nelson, Herman Nelson Corp., Moline, Ill., kindly loaned the four fan units used in the field test and provided technical advice. E. DeCamp and M. Terrill, Philip Carey Co., Lockland, Ohio, gave much of their own time and effort in installing the ducts used in the laboratory and field tests. J. E. Kearns, General Electric Co., assisted in the design and manufacture of our improved vaporizer. This paper owes much to the technical assistance of Margaret Mellody and Silva Trautman, who carried out the bacteriologic and chemical work.

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Periodic Heat Flow —Homogeneous Walls or Roofs

By C. O. Mackey* and L. T. Wright, Jr.,** Ithaca, N. Y.

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers in cooperation with Cornell University.

SUMMARY—The results of a cooperative research program obtained during 1943 are presented in this paper. An exact and an approximate method are given for finding the contribution to the cooling load due to heat transfer from the inside surface of a homogeneous wall or roof under the following conditions: (1) a cyclic variation of the temperature of outdoor air and the total solar radiation incident upon the building surface with a period of 24 hours; (2) a constant temperature of the indoor air. More complete preliminary data were presented to the Research Technical Advisory Committee on Cooling Load in Summer Air Conditioning.

THIS REPORT summarizes the work done during 1943 on unsteady heat flow through homogeneous building materials. The general method followed is that explained in a previously published paper,[1] but a brief summary of the method is included in this report. Ranges chosen for the variables are those suggested by the ASHVE Research Technical Advisory Committee on Cooling Load in Summer Air Conditioning. More complete results of the study have been presented to this Committee in the form of preliminary reports.

Variables, Symbols, and Units

The variables involved in this study of periodic heat flow are listed in Table 1 with their symbols, units, and suggested ranges.

Sol-Air Temperature[2]

In the first paper by Mackey and Wright,[3] a new term was introduced which is important in the study of weather data and solar heat gain—*equivalent temperature* of outdoor air. Since the term *equivalent temperature* has been widely used for many years in England to give a basis for estimating comfort conditions the authors accepted the suggestion of an alternative term *sol-air temperature* and have used it throughout this paper. Sol-air temperature will be understood to mean the temperature of the outdoor air, which in contact with a shaded building surface, would give the same rate of heat transfer and the same temperature distribution through the material as exists with the actual dry-bulb temperature of the outdoor air and the actual intensity of solar radiation incident upon that surface. For either steady or unsteady flow of heat, the sol-air temperature for a value of the outdoor air film coefficient of heat transfer of 4 Btu/hr ft² F is:

$$t_e = t_a + \frac{bI}{4} \quad \ldots\ldots\ldots\ldots\ldots\ldots (1)$$

where

t_a = the dry-bulb temperature of the outdoor air, Fahrenheit.

I = the intensity of solar radiation incident upon the outdoor surface, Btu/hr ft.²

For example, if the intensity of solar radiation incident upon a building surface with a solar absorptivity of 0.4 is 200 Btu/hr ft,² the effect of the solar radiation upon the rate of heat transfer through the material and upon the distribution of temperature in that material is precisely as if the temperature of the outdoor air were 20 F higher with the surface receiving no solar radiation. In this example, the sol-air temperature would be 20 F higher than the shaded dry-bulb temperature of the air.

The equation which fits the periodic sol-air temperature as a function of time is of the general form:

$$t_e = t_m + \sum_{n=1}^{\infty} t_n \cos\,(15n\,\theta - a_n) \ldots (2)$$

where

t_m = the daily average sol-air temperature, F

θ = the time, measured in hours after noon (at noon, $\theta = 0$; at 1 p.m., $\theta = 1$, etc.)

n = the harmonic coefficient ($n = 1$ for first harmonic, or fundamental; $n = 2$ for second harmonic, etc.)

t_n = the harmonic temperature coefficient, Fahrenheit (t_1 for first harmonic, etc.)

a_n = the harmonic phase angle, degrees, (a_1 for first harmonic, etc.)

Temperature of Inside Surface of Building

The exact solution is complete for finding the temperature of the inside surface of a *homogeneous building material* at any time for the case where (1) the incident

Table 1—Variables, Symbols, Units and Suggested Ranges

Quantity	Units	Symbol	Range Suggested by Committee
Inside air film coeff. of heat transfer	Btu/hr ft² F	h_o	1.65 (fixed)
Outdoor air film coeff. of heat transfer	Btu/hr ft² F	h_L	4.00 (fixed)
Thermal conductivity	Btu/hr ft F	k	0.00833 to 28
Volumetric specific heat	Btu/ft³ F	ρc	1 to 60
Thickness of material	ft	L	0.0104 to 3
Absorptivity of outside surface for solar radiation	none	b	0; 0.4; 0.7 (selected values)

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[1] Summer Comfort Factors as Influenced by the Thermal Properties of Building Materials, by C. O. Mackey and L. T. Wright, Jr. (ASHVE Journal Section, *Heating, Piping & Air Conditioning*, December 1942.)

[2] This discussion applies only to those building materials which do not directly transmit solar radiation.

[3] Loc. Cit. See Note 1.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Grand Rapids, Mich., June 1944.

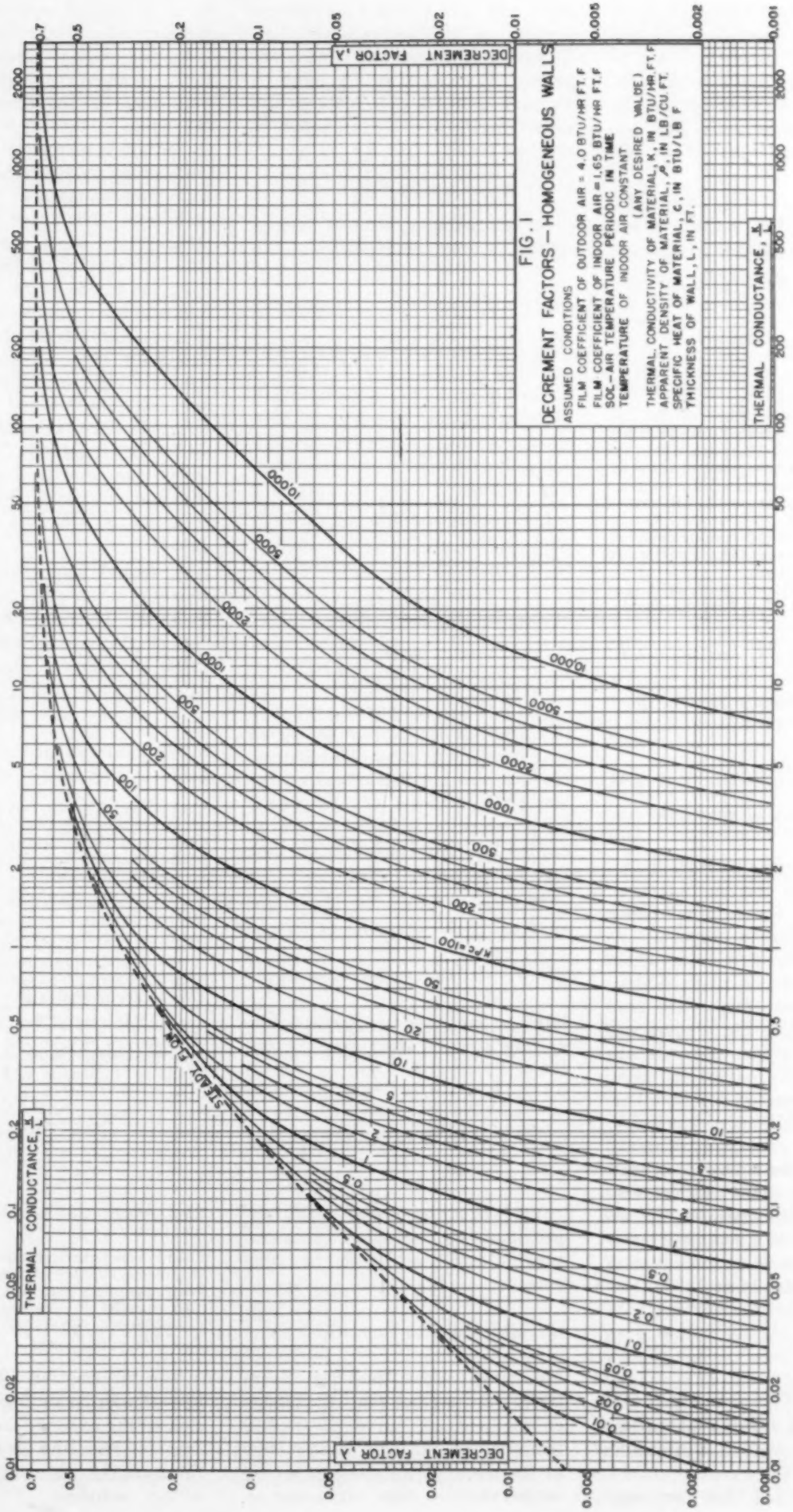


FIG. 1
DECREMENT FACTORS — HOMOGENEOUS WALLS
ASSUMED CONDITIONS
FILM COEFFICIENT OF OUTDOOR AIR = 4.0 BTU/HR FT.F
FILM COEFFICIENT OF INDOOR AIR = 1.65 BTU/HR FT.F
SOL.-AIR TEMPERATURE PERIODIC IN TIME
TEMPERATURE OF INDOOR AIR CONSTANT
(ANY DESIRED VALUE)
THERMAL CONDUCTIVITY OF MATERIAL, K, IN BTU/HR FT.F
APPARENT DENSITY OF MATERIAL, ρ, IN LB/CU FT.
SPECIFIC HEAT OF MATERIAL, C, IN BTU/LB F
THICKNESS OF WALL, L, IN FT.
DECREMENT FACTOR, λ
THERMAL CONDUCTANCE, K/L
STEADY FLOW
RPE=100

solar radiation and temperature of outdoor air are periodic, (2) the temperature of the indoor air is constant, (3) the outdoor air film coefficient of heat transfer is 4.0, and (4) the indoor air film coefficient of heat transfer is 1.65 Btu/hr ft² F.

For this case, at any time θ, the temperature of the inside surface of the building material is:

$$t_o = t_i + \frac{0.606\,(t_m - t_i)}{0.856 + \frac{L}{k}} + \sum_{n=1}^{\infty} \lambda_n t_n \cos\,(15n\,\theta - a_n - \phi_n) \quad \ldots\ldots\ldots (3)$$

where

t_o = the temperature of the inside surface of the building material, Fahrenheit
t_i = the constant temperature of the indoor air, Fahrenheit
L = the thickness of the material, ft
k = the thermal conductivity of the material, Btu/hr ft Fahrenheit
λ_n = the harmonic decrement factor (λ_1 for first harmonic, etc.)
ϕ_n = the harmonic lag angle, degrees (ϕ_1 for first harmonic, etc.)

The instantaneous rate of heat transfer from the indoor surface, in Btu/hr, is:

$$q = 1.65\,A\,(t_o - t_i) \quad \ldots\ldots\ldots (4)$$

where

A = the area of the inside surface at temperature t_o, ft.²

Equations for the decrement factor, λ, and for the lag angle, ϕ, were given in the original paper and are repeated in Appendix A. Graphs are presented from which these values may be read. Values of the decrement factor are read from Fig. 1, while Fig. 2 gives values of the lag angle. For the nth harmonic of the *actual* material, use the fundamental (or first harmonic) of a fictitious or *equivalent* material; the equivalent material designated by the subscript e, must then have the same thermal conductivity and thickness as the actual material but a volumetric specific heat which is n times that of the actual material; or

$$\left(\frac{k}{L}\right)_e = \frac{k}{L} \quad \ldots\ldots\ldots (5)$$

while

$$(k\rho c)_e = nk\rho c \quad \ldots\ldots\ldots (6)$$

A numerical illustration of the exact solution is given in Appendix B.

It is recognized that few engineers will care to use a method as complex as the accurate solution. An approximate method is next explained which is simple enough to warrant general use without too great sacrifice in the accuracy of the final result.

Approximate Solution

An approximate solution is presented for the temperature of the inside surface of the material which gives acceptable results and which starts with a knowledge of the periodic sol-air temperature as a function of time.

Let the steady-flow mean daily temperature of the inside surface of the material be t_M, where

$$t_M = t_i + \frac{0.606\,(t_m - t_i)}{0.856 + \frac{L}{k}} \quad \ldots (7)$$

With two assumptions explained in Appendix C, it may be shown that Equation 8 gives the temperature of the inside surface of the material in terms of the sol-air temperature at a time which is earlier by the fundamental time lag, in hours, $(\phi_1/15)$.

The temperature of the inside surface of the material at a time $(\theta + \frac{\phi_1}{15})$ hours after noon is related to the sol-air temperature at a time θ hours after noon as follows:

$$(t_o)_{\theta + \frac{\phi_1}{15}} = t_M + \lambda_1\,[(t_o)_\theta - t_m] \quad \ldots\ldots\ldots (8)$$

This equation states that (approximately) the sol-air temperature at any time affects the temperature of the inside surface of the material at a time which is $\frac{\phi_1}{15}$ (the fundamental time lag) hours later. It implies that the shape of the periodic curve of surface temperature *vs.* time is the same as the shape of the periodic sol-air temperature *vs.* time curve, but reduced in range and displaced in phase.

In preliminary reports, the results obtained for the variation with time of the temperature of the inside surface of widely different building materials in different thicknesses by both the exact and approximate methods have been compared. The shape of the temperature *vs.* time curve and the time of maximum heat transfer from the inside surface have been found to be reproduced fairly well by the approximate method. The maximum rate of heat transfer found by the approximate method has always been found to be on the safe side—greater than that given by the exact method. The greatest percentage error in the heat transfer rate has always been found to occur at the times of the smallest rates.

A second approximate method which gives more accurate results than the method just explained, but at the cost of a little more work, is presented in Appendix C.

Summary of Results

The assumptions that have been made in this study follow:

(1) The temperature of the outdoor air and the total solar radiation incident upon the building surface have been assumed to be cyclic with a period of 24 hours.

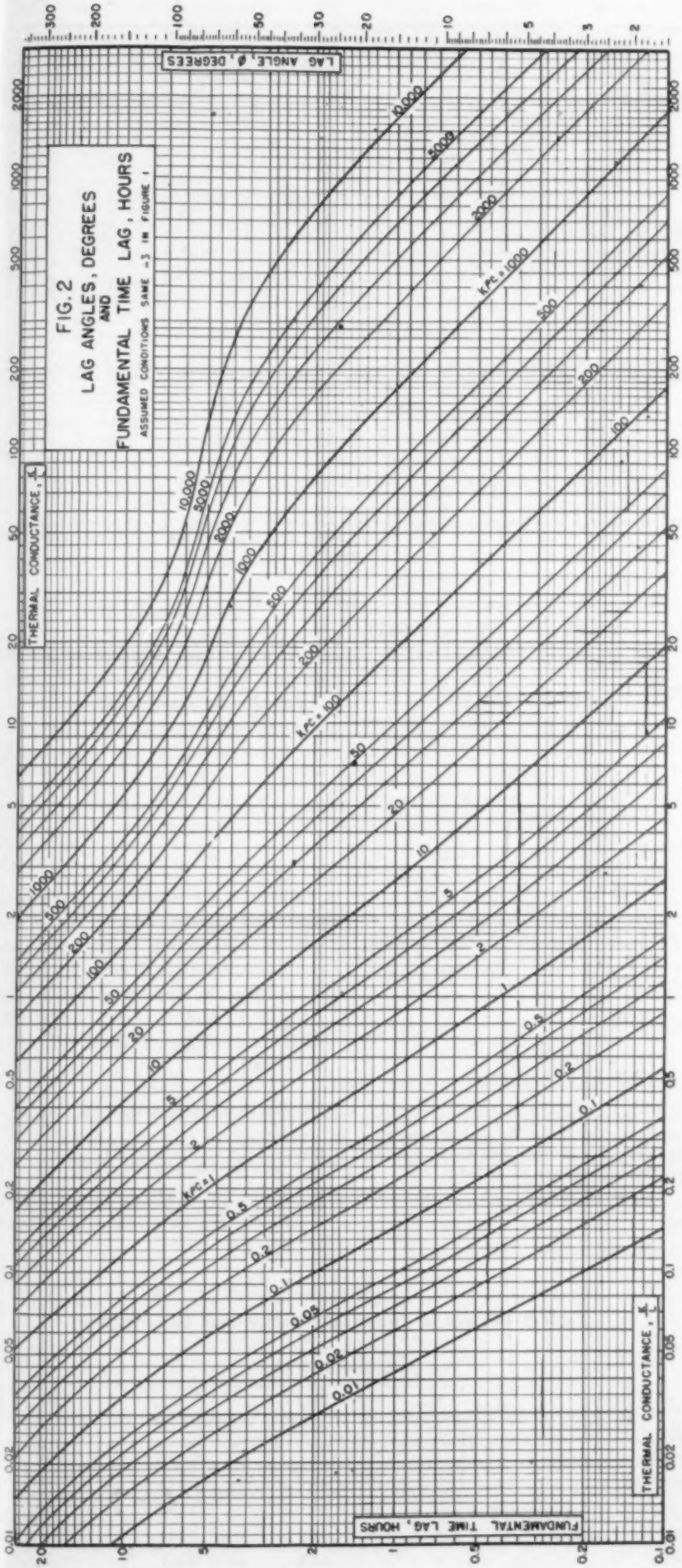
(2) The temperature of the indoor air has been assumed to be held constant during the period of 24 hours.

(3) The building wall or roof has been assumed to be made of a single, homogeneous material.

(4) The rate of heat transfer from the outdoor air to the building surface has been assumed constant and equal to 4 Btu/hr ft²F (outside air film coefficient of heat transfer; summer conditions).

(5) The rate of heat transfer from the inside surface of the structure has been assumed constant and equal to 1.65 Btu/hr ft² for each degree of temperature difference between surface and indoor air.

The information which must be known in order to find the contribution to the cooling load due to heat transfer from the inside surface of the building material at any time of day includes:



(1) The periodic sol-air temperature *vs.* time curves for the particular locality, orientation and solar absorptivity of the building surface (design curves).

(2) The thickness, thermal conductivity, and volumetric specific heat of the wall or roof and the solar absorptivity and orientation of the building surface.

(3) The constant temperature of the indoor air.

Example of Recommended Procedure (Approximate Method of Solution)

(1) Assume the sol-air temperature *vs.* time curve for the locality under consideration to be given by Fig. 3. (These design curves should be based upon weather data.)

(2) Assume the unshaded wall under consideration faces west and is made of homogeneous red brick with a thickness of 8 in., a thermal conductivity of 5 Btu in./hr ft²F, a volumetric specific heat of 20 Btu/ft³F, and a solar absorptivity of the exterior surface of $b = 0.7$.

(3) Assume the constant temperature of the indoor air to be 80 F.

It is required to find the maximum contribution to the cooling load due to heat transfer from the inside surface of this wall, the time of this maximum, and the contribution to the cooling load at a time of 3 p.m. (Actually, the method permits finding the rate of this heat transfer at any specified time.)

Steps in solution:

(1) From Fig. 1, for $\frac{k}{L} = 0.625$ and $k\rho c = 8.33$, read the fundamental decrement factor, λ_1, to be 0.15.

(2) From Fig. 2, for $\frac{k}{L} = 0.625$ and $k\rho c = 8.33$, read the fundamental time lag to be 5.5 hour.

(3) From Fig. 3, the maximum sol-air temperature for a wall facing west with solar absorptivity of 0.7 is 133.2 F, and occurs 3.7 hours after noon.

The maximum contribution to the cooling load from heat transfer through this wall will then occur at (3.7 + 5.5) or 9.2 hours after noon (9:12 p.m.). At this time, the temperature of the inside surface is found from Equation 8. The average daily sol-air temperature (Fig. 3) for the west wall, $b = 0.7$, is $t_m = 91.6$ F. The steady-flow mean inside surface temperature (corresponding to an outdoor air temperature of 91.6 F and an indoor air temperature of 80 F) is, from Equation 7,

$$t_M = 80 + \frac{0.606\ (11.6)}{0.856 + 1.6} = 82.9\ \text{F}$$

Then, from Equation 8, the temperature of the inside surface of the wall at 9:12 p.m. is:

$$t_o = 82.9 + 0.15\ (133.2 - 91.6) = 89.1\ \text{F}$$

The corresponding maximum rate of heat transfer from the inside surface of the wall, from Equation 4, is:

$$\frac{q}{A} = 1.65\ (89.1 - 80) = 15.0\ \text{Btu/hr ft}^2$$

The next part of the example is to find the rate of heat transfer at 3 p.m. The sol-air temperature which influences this rate of heat flow is that temperature at a time 5.5 hours (fundamental time lag) earlier than 3 p.m. or at 2.5 hours before noon (9:30 a.m.). From Fig. 3, the sol-air temperature at this time is found to be 87.8 F. Then, from Equation 8, at 3 p.m., the temperature of the inside surface of the wall is:

$$t_o = 82.9 + 0.15\ (87.8 - 91.6) = 82.3\ \text{F}$$

The corresponding contribution to the cooling load at 3 p.m. is:

$$\frac{q}{A} = 1.65\,(82.3 - 80) = 3.8\ \text{Btu/hr ft}^2$$

It will be noted that this recommended solution uses the approximate method and that the method will give the rate of heat transfer at any specified time of day.

Effect of Values of Air Film Coefficients

Although Figs. 1 and 2 are *exactly* correct only when the outdoor air film coefficient of heat transfer is 4.0 and the indoor air film coefficient of heat transfer is 1.65 Btu/hr ft² F, values of the decrement factor and lag angle read from these graphs are approximately correct for the usual departures from these values. This is discussed further in Appendix D.

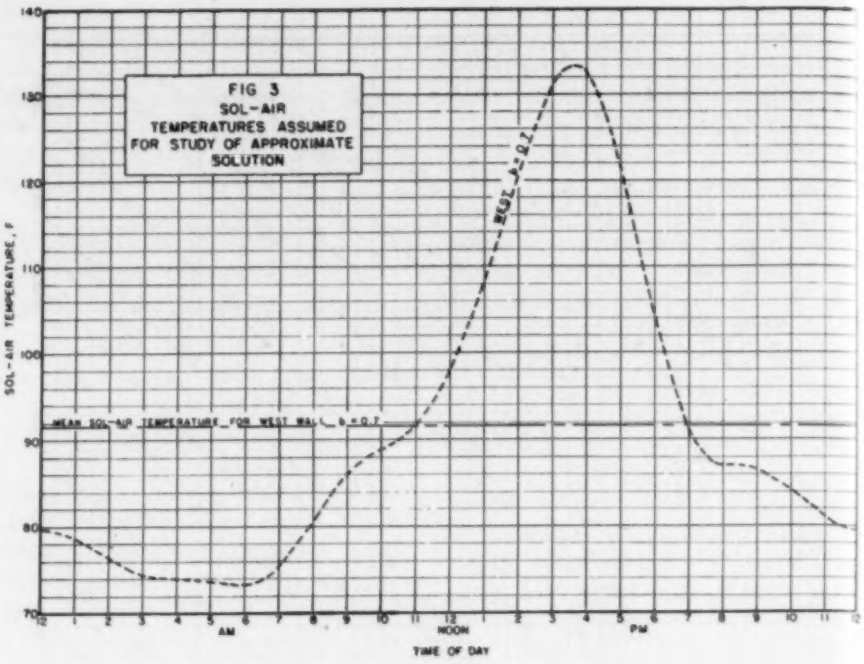
Appendix A

Equations for Decrement Factor λ and Lag Angle φ.

Decrement factor:

$$\lambda = \sqrt{\frac{2}{F^2 + G^2}} \qquad \text{.....(A1)}$$

Lag Angle



$$\phi = \tan^{-1}\left(\frac{F - G}{F + G}\right) \qquad \text{.....(A2)}$$

where

$$F = (\pi_1 + 1)\ C_1 + \frac{C_3}{\pi_3} + 2\pi_1\pi_2 C_4$$

$$G = (\pi_1 + 1)\ C_2 + \frac{C_4}{\pi_3} - 2\pi_1\pi_2 C_3$$

$$C_1 = \cos\pi_2 \cosh\pi_2 + \sin\pi_2 \sinh\pi_2$$

$$C_2 = \cos\pi_2 \cosh\pi_2 - \sin\pi_2 \sinh\pi_2$$

$$C_3 = \sin\pi_2 \cosh\pi_2$$

$$C_4 = \cos\pi_2 \sinh\pi_2$$

$$\pi_1 = \frac{h_o}{h_L}$$ (fixed at 0.4125 for Figs. 1 and 2)

$$\pi_2 = \frac{sL}{k_s}$$

$$\pi_3 = \frac{}{h_o}$$

$$s = \sqrt{\frac{0.1309\rho c}{k}}$$

The decrement factors and lag angles given in Figs. 1 and 2 are the fundamental or first harmonic values. For the harmonics of order n, use

$$\pi_{2_n} = \sqrt{n}\ \pi_{2_1},$$

and

$$\pi_{3_n} = \sqrt{n}\ \pi_{3_1}.$$

Appendix B

A numerical illustration of the exact method of solution follows:

Assumed sol-air temperature series for west wall ($b = 0.7$):

$$t_o = 91.6 + 23\cos(15\theta - 51) + 9.1\cos(30\theta - 92) + 5.9\cos(45\theta - 166) + 3.6\cos(60\theta - 217) + 1.0\cos(75\theta - 292) + 0.93\cos(90\theta - 68) \qquad \text{.....(A3)}$$

Assume an unshaded west wall made of homogeneous red brick with a thickness of 8 in., a thermal conductivity of 5 Btu in./hr ft² F, a volumetric specific heat of 20 Btu/ft³ F, and a solar absorptivity of the exterior surface of 0.7. For this wall,

$$\frac{k}{L} = 0.625\ \text{Btu/hr ft}^2\text{F},$$

and

$$k\rho c = 8.33\ \text{Btu}^2\text{/hr ft}^4\text{F}^2$$

Values of the decrement factor λ and of the lag angle φ are shown in Table 2.

Table 2—Values of Decrement Factor and Lag Angle

		k/L = 0.625		
Harmonic	n	$(k\rho c)_o$	λ_n	ϕ_n
First (fundamental)	1	8.33	0.150	83
Second	2	16.7	0.080	131
Third	3	25.0	0.048	166
Fourth	4	33.3	0.030	195
Fifth	5	41.7	0.020	220
Sixth	6	50.0	0.014	242

For an assumed constant temperature of the indoor air of $t_i = 80$ F, the temperature of the inside surface of the wall at any time θ is found from Equation 3, Equation A3, and these tabular values as:

$$t_o = 80 + 2.86 + 3.45 \cos (15\,\theta - 134) + 0.729 \cos (30\,\theta - 223) + 0.283 \cos (45\,\theta - 332) + 0.108 \cos (60\,\theta - 412) + 0.020 \cos (75\,\theta - 512) + 0.013 \cos (90\,\theta - 310)$$

At a time of 8 p.m. ($\theta = 8$), for example, the temperature of the inside surface of this wall is:

$$t_o = 80 + 2.86 + 3.35 + 0.70 + 0.25 + 0.04 + 0.00 + 0.01 = 87.21 \text{ F}$$

At this time, the instantaneous rate of heat transfer from the inside, or room surface, of this wall is found from Equation 4 to be:

$$\frac{q}{A} = 1.65\ (87.21 - 80) = 11.90 \text{ Btu/hr ft}^2.$$

The approximate method of solution yields a result for the maximum rate of heat transfer from the inside surface of this wall which is about 25 per cent higher than the actual maximum rate, a maximum temperature of the inside surface of the wall about 1.9 F higher than the actual maximum temperature, and a time of this maximum about one hour later than the time of the actual maximum. The same curve for sol-air temperature (Fig. 3) is assumed for both cases.

Appendix C

The actual temperature of the inside surface of the material at a time θ hours after noon is:

$$(t_o)_\theta = t_M + \sum_{n=1}^{\infty} \lambda_n t_n \cos (15n\,\theta - a_n - \phi_n) \quad \text{.....(C1)}$$

The fundamental time lag, in hours, is the fundamental lag angle divided by 15; the temperature of the inside surface of the material, at a time $\theta + \frac{\phi_1}{15}$ hours after noon is:

$$(t_o)_{\theta + \frac{\phi_1}{15}} = t_M + \sum_{n=1}^{\infty} \lambda_n t_n \cos [15n\,(\theta + \phi_1) - a_n - \phi_n]$$

$$= t_M + \sum_{n=1}^{\infty} \lambda_n t_n \cos [15n\,\theta - a_n - (\phi_n - n\,\phi_1)] \quad \text{.....(C2)}$$

The first assumption made is that:

$$\phi_n = n\,\phi_1 \quad \text{.....(C3)}$$

The second assumption made is that:

$$(1 - \lambda_n) = (1 - \lambda_1) \quad \text{.....(C4)}$$

With these two assumptions, it may be shown that the temperature of the inside surface of the material at a time $\theta + \frac{\phi_1}{15}$ hours after noon is related to the sol-air temperature at a time θ hours after noon as follows:

$$(t_o)_{\theta + \frac{\phi_1}{15}} = t_M + \lambda_1 [(t_o)_\theta - t_m] \quad \text{.....(C5)}$$

The reasonableness of the two assumptions made in the approximate solution may be examined. The first assumption is that each harmonic lag angle is the fundamental lag angle multiplied by the order of the harmonic. In the example given in explanation of the exact solution, the fundamental lag angle was 83 deg, the second harmonic lag angle 131 deg, the third harmonic 166 deg, etc.; this assumption would be true if the second harmonic lag angle were 166 deg, the third harmonic 249 deg, etc. For materials of low volumetric specific heat, this assumption is closely met.

The second assumption states, in effect, that all decrement factors, regardless of the order of the harmonic, are equal for a given material. This is far from true, except for materials having a low volumetric specific heat. Any error introduced by this assumption is reduced by the fact that the successive harmonic temperature coefficients in the equation for sol-air temperature (the t_n's in Equation 2) decrease quite rapidly in the usual case. This means that the values of the successive terms in the series solution for the temperature of the inside surface of the building material drop off rapidly; this fact is illustrated in the one example given previously. Also, for walls with a high volumetric specific heat, the fundamental decrement factor (λ_1) is quite small, the second harmonic decrement factor (λ_2) is still smaller, etc.; in this case, $(1 - \lambda_1)$ is a good approximation of $(1 - \lambda_n)$.

Another approximate method is next presented which gives more accurate values but requires slightly more work. The following equation gives the temperature of the inside surface of the material:

$$(t_o)_{\theta + \frac{\phi_1}{15}} = t_M + \lambda_e [(t_o)_\theta - t_m] \quad \text{.....(C6)}$$

where

$$\lambda_e = \frac{\lambda_1 + \lambda_2}{2} \quad \text{.....(C7)}$$

It will be noted that the only difference between Equation C6 and Equation C5 is the decrement factor, λ, used. In Equation C5, the fundamental (or first harmonic) decrement factor is used, while in Equation C6, the arithmetic mean of the fundamental and second harmonic decrement factor is used.

When this latter method is followed, the fundamental decrement factor is read from Fig. 1 when the values of k/L and $k\rho c$ are known for the actual material and thickness; the second harmonic decrement factor is then read from the same figure for the same value of k/L, but the equivalent value of $k\rho c$ which is twice the value of $k\rho c$ for the actual material (see Equations 5 and 6). For the 8-in. brick wall, these decrement factors are $\lambda_1 = 0.15$ and $\lambda_2 = 0.08$. By using $\lambda_e = 0.115$, a more accurate estimate of the rate of heat transfer from the inside surface may be obtained. For the same data as used in the recommended procedure, the approximate maximum temperature of the inside surface of the brick wall facing west is, (from Equation C6),

$$t_o = 82.9 + 0.115\ (133.2 - 91.6) = 87.7 \text{ F}$$

The corresponding maximum rate of heat transfer from the inside surface of this wall is:

$$\frac{q}{A} = 1.65\ (87.7 - 80) = 12.7 \text{ Btu/hr ft}^2$$

This result is closer to the exact series solution of Appendix B which gave 11.9 Btu/hr ft² for this rate than the simpler approximate method which gave a result of 15.0 Btu/hr ft².

Appendix D

The effect of using values of the outdoor and indoor film coefficients of heat transfer other than the assumed values of 4 and 1.65 Btu/hr ft² F, respectively, may be demonstrated by examples. Calculated values of the fundamental decrement factor, the fundamental lag angle, and the fundamental time lag are shown in Table 3 for the 8-in. brick wall of the previous examples (k = 0.417 Btu/hr ft F; ρc = 20 Btu/ft³ F).

With the outdoor air film coefficient of heat transfer held constant, a decrease in the value of the indoor air film coefficient slightly increases the time lag and appreciably increases the fundamental decrement factor. The effect is to increase the daily range in temperature of the inside surface; the maximum temperature of that surface is raised, and the minimum temperature is lowered. With the indoor air film coefficient of heat transfer held constant, a decrease in the value of the outdoor air film coefficient slightly increases the time lag but decreases the fundamental decrement factor. The effect is to decrease the daily range in temperature of the inside surface; the maximum temperature of that surface is lowered and the minimum temperature raised. At the same time, any departure of the outdoor air film coefficient from the assumed value of four changes the sol-air temperature at any time when sun is shining on the building surface; a decrease in this coefficient will raise the sol-air temperature.

Table 3—Calculated Values of Fundamental Decrement Factor, Lag Angle, and Time Lag for 8-Inch Brick Wall

Outdoor Air Film Coeff. of Heat Transfer	Indoor Air Film Coeff. of Heat Transfer	Fundamental Decrement Factor	Fundamental Lag Angle, degrees	Fundamental Time Lag, hr
h_L	h_o	λ_1	ϕ_1	$\phi_1/15$
4.0	1.65	0.150	83	5.5
4.0	0.50	0.234	97	6.5
4.0	1.50	0.158	84	5.6
3.0	1.50	0.146	87	5.8
2.0	1.50	0.126	92	6.1

Table 4—Effect of Film Coefficient Changes Upon Maximum Rate of Heat Transfer From Inside Surface

Outdoor Air Film Coeff. of Heat Transfer	Indoor Air Film Coeff. of Heat Transfer	Maximum Rate of Heat Transfer from Inside Surface, (approximate method) Btu/hr ft²	Time of Maximum Rate of Heat Transfer (approximate method)
h_L	h_o	q/A	p. m.
4.0	1.65	15.0	9:12
4.0	0.50	7.9	10:12
4.0	1.50	14.5	9:18
3.0	1.50	16.9	9:20
2.0	1.50	20.6	9:48

The complete effect of these changes in film coefficients upon the maximum rate of heat transfer from the inside surface of the 8-inch brick wall facing west is shown in Table 4.

The *indoor* air film coefficient of heat transfer is a combined coefficient of convective and radiant heat transfer. For large vertical surfaces, the film coefficient of heat transfer by natural convection is 0.27 $(\triangle t)^{0.25}$; for a temperature difference of 10 F, this coefficient is 0.48 Btu/hr ft² F. For warm horizontal surfaces facing down, this film coefficient is 0.2 $(\triangle t)^{0.25}$ or 0.16 Btu/hr ft² F for a temperature difference of 10 F. The radiant heat exchange at the interior surface is complicated; for an interior surface with an emissivity of unity which exchanges heat with other *black* surfaces at *air temperature*, the rate of radiant heat loss from the warm surface is, closely, 1.1 Btu/hr ft² for each degree of temperature difference from surface to air. If all the other interior surfaces of an enclosure are at the same temperature as the inside surface of an exterior wall or roof, or if the inside surface of this exterior wall or roof is covered with a material having an emissivity of zero, there can be no radiant heat loss from that surface. With common nonreflecting inside surfaces, the combined film coefficient of heat transfer for vertical walls in still air is probably between 1.5 and 1.6 Btu/hr ft² F, provided that the temperature of all other surfaces in the enclosure is the same as the temperature of the air; for horizontal roofs under similar conditions, this combined film coefficient is slightly lower. In this report, Figs. 1 and 2 are based upon values of the outdoor and indoor air film coefficients of heat transfer of 4.0 and 1.65 Btu/hr ft² F, respectively. As far as the indoor film coefficient of heat transfer is concerned, it is believed that these decrement factors and lag angles are sufficiently exact for non-reflecting inside surfaces *seeing* other interior surfaces at air temperature. In support of this contention, note that with the 8-in. brick wall of the example, a change in the value of h_o from 1.65 to 1.50 lowers the maximum rate of heat transfer at the inside surface from 15.0 to 14.5 Btu/hr ft.² On the other hand, these figures would not give correct results for exterior walls or roofs lined with a highly reflecting substance or for a case where the other interior surfaces of the enclosure were at the same temperature as the surface in question. Note that a highly reflective lining applied to the inside surface of the brick wall would lower the maximum rate of heat transfer from 15.0 to 7.9 Btu/hr ft.²

The value of the outdoor air film coefficient of heat transfer has a multiple effect in the problem. It affects the sol-air temperature and the values of the decrement factor and lag angle. Equation 1 for sol-air temperature is admittedly imperfect in some respects. During the day, the wind velocity changes and the value of the outdoor air

film coefficient of heat transfer also changes. Instead of a constant value of h_L of 4 Btu/hr ft² F, a value that depended upon the instantaneous wind velocity should be used for perfection. During clear nights, there is a considerable loss of heat from the exterior wall by radiation to the cold sky; this effect, which would tend to lower the daily mean sol-air temperature, is not considered, but its neglect is on the safe side. Values of the outdoor air film coefficient of heat transfer are commonly based upon the cooperative research[4] between the University of Minnesota and the ASHVE. Although these tests did not duplicate outdoor air conditions, exactly, they showed that the film coefficient increased with air movement, mean temperature and surface roughness. For a brick surface at a mean temperature of 80 F, for example, values of the film coefficient were found to be about 2 for a parallel air velocity of 0 mph, 4 for an air velocity of 5 mph, and 6 for an air velocity of 10 mph. The last table shows that a decrease in h_L from 4 to 2 Btu/hr ft² F, through the combined effects on sol-air temperature and decrement factor will raise the maximum rate of heat transfer from the inside surface of the 8-in. brick wall from 14.5 to 20.6 Btu/hr ft² for $h_o = 1.5$, when the result is obtained by the approximate method of solution; also, the time lag is increased by about 0.5 hour. It should be noted, however, that the value of 15 Btu/hr ft² for the maximum rate of heat transfer as found from the *approximate* method, is roughly, 25 per cent higher than the value of 11.9 found by the exact but complex series method in Appendix B for the same wall. This exact series method applied with sol-air temperature and decrement factor corrected to $h_L = 2$ would give about the same result as the approximate method with $h_L = 4$.

The authors believe that the simplest procedure is to use a constant value of h_L of 4 in the equation for sol-air temperature, regardless of the wind velocity. Then, by using the approximate method to find the maximum rate of heat transfer based upon Figs. 1 and 2 for $h_L = 4$ and $h_o = 1.65$, the results will be very close for the lowest possible outdoor air movement; for a wind velocity of 10 mph, the estimated rate of heat transfer is always on the safe side. Further, the approximate method always gives a slightly greater time lag than the exact method, and the effect of decreased outdoor air movement is in the same direction.

[4]Surface Conductances as Affected by Air Velocity, Temperature and Character of Surface, by F. B. Rowley, A. B. Algren, and J. L. Blackshaw. (ASHVE Transactions. Vol. 36, 1930, p. 429.)

Discussion

F. E. Giesecke, College Station, Tex. (Written): The authors have earned the gratitude of the engineering profession by their splendid work, reflected by this paper and by the paper presented at the January 1943 meeting.[5] Their work will, no doubt, be continued so as to make the results of their studies of greater value to the practicing engineer.

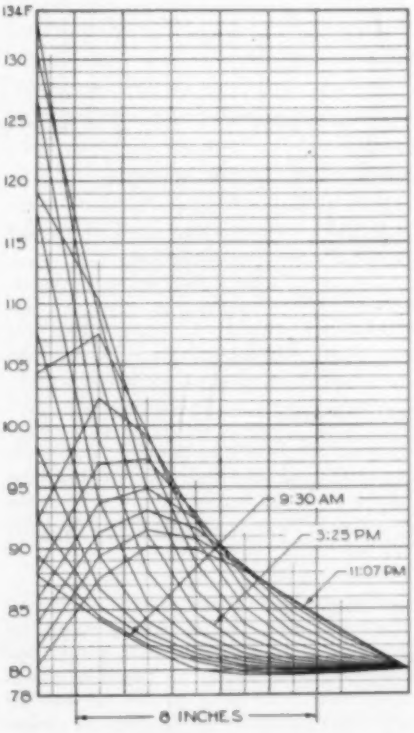


Fig. A—Graphical analysis of heat flow through brick wall

The following brief discussion and the experimental data submitted herewith may be of some assistance in future studies.

If the flow of heat through the 8-in. brick wall, described in the paper, is studied by means of the approximate graphical method, described in the ASHVE Transactions, Vol. 45, 1939, pp. 441-458, the results will be as shown in Fig. A. To employ graphical analysis, it is necessary to know or to assume the thermal gradient in the wall at the beginning of the study. In this case, it was assumed that the interior surface of the wall had cooled slightly below 80 F during the night and that by 9:30 a. m. the indoor air temperature had risen to 80 F, so that, at the beginning of the study, there was still a slight flow of heat into the 8-in. wall from the indoor air. It was also assumed that the outdoor sol-air temperature was 87.8 F at 9:30 a. m. and that the form of the thermal gradient at 9:30 a. m. was similar to the forms of the thermal gradients shown in Fig. 13 of the Transactions article[6], which thermal gradients had been determined by actual measurements.

[5]Summer Comfort Factors as Influenced by Thermal Properties of Building Materials, by C. O. Mackey and L. T. Wright, Jr. (ASHVE Journal Section, *Heating, Piping & Air Conditioning*, December 1942.)

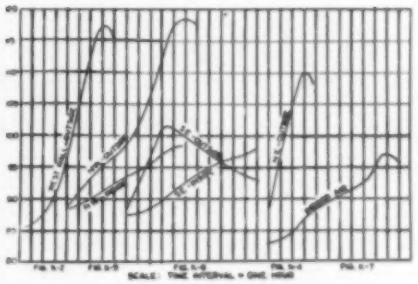


Fig. B—Variation in surface temperatures of 8-in. wall

To study the flow of heat, the 8-in. wall was divided into five slabs, each 1.6 in. thick. The corresponding time increment was calculated to be 51.3 min for the physical qualities of the brick wall specified in the paper. Having determined the time increment and knowing the varying sol-air temperature (Fig. 3) the successive sol-air temperatures were found and the 17 successive thermal gradients drawn in the usual manner. The first thermal gradient shown in Fig. A is for 9:30 a. m. and the last one for 11:07 p. m.

It appears from the diagram that the interior wall surface temperature had risen from about 79.6 F to about 80.5 F or about 0.9 F during the time interval from 9:30 a. m. to 3:25 p. m. This would correspond to a rate of heat flow into the room of about 1.48 Btu per hour if the first thermal gradient had been assumed on the basis of an interior wall surface temperature of 80 F at 9:30 a. m. instead of 79.6 F.

It also is apparent from the dia-

[6]Loc. cit. See Note 5, p. 455.

Fig. C—Building where heat flow studies were made

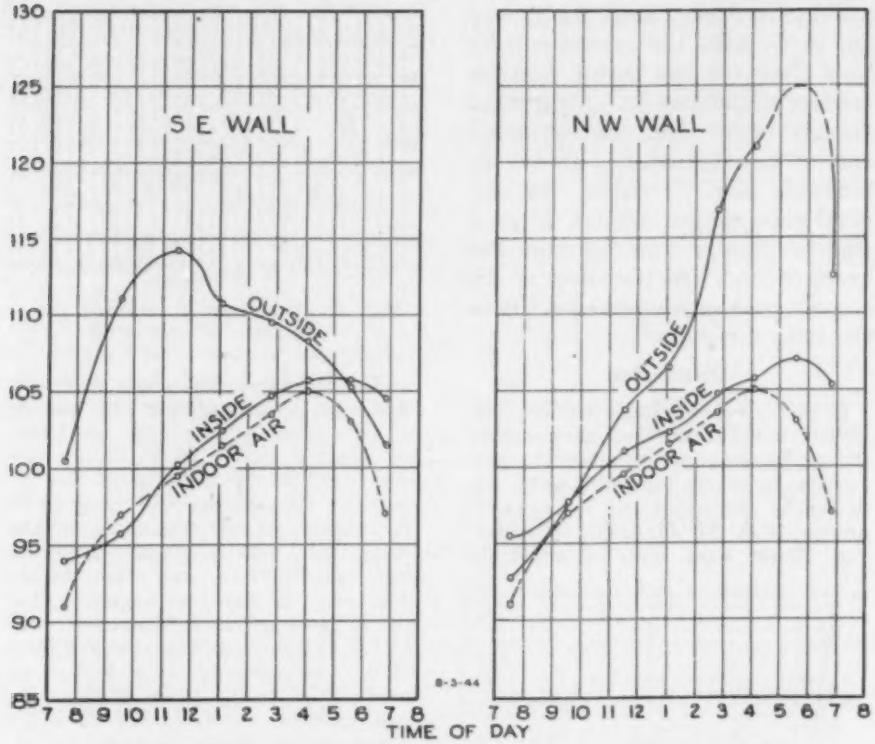


Fig. D—Surface temperature charts for S. E. and N. W. walls

gram that the maximum rate of heat flow into the room occurred at about 11:30 p. m. and that it is at the rate of about 1.65 (84.5-80.0) or 7.44 Btu per hour; however, the increase in heat flow from 9:12 p. m. to 11:30 p. m. is very slight, according to the graphical analysis. These values are somewhat lower than those calculated by the authors; the differences are, no doubt, the results of differences in the methods of analysis.

It also appears from this diagram that the exterior wall surface temperatures of the author's 8-in. wall are as shown in Fig. B (x-2) for the sol-air temperatures shown in Fig. 3 and for the rates of heat flow shown in Fig. A.

To secure a few experimental data, readings were taken on June 14, 1944 on three of the 8-in. red brick walls of the second story of the building whose east corner is shown in Fig. C. This building fronts S, 38 deg 30 min E. As there were clouds in the sky during part of the time the temperatures recorded are somewhat lower than would be obtained under complete sunshine conditions.

The varying exterior wall surface temperature of the northeast wall of this building is shown in Fig. B (x-4); the varying exterior and interior wall surface temperatures of the northwest wall are also shown in Fig. B (x-5). The varying exterior and interior wall surface temperatures of the southeast wall are shown in Fig. B (x-6). The varying indoor air temperature of the space between the northwest and southeast walls is shown in Fig. B (x-7). This space receives its heat primarily through the roof. The temperature of the southwest wall was not taken because the wall is in the shade of another building.

An opportunity to obtain temperatures on a clear warm day was afforded on August 4, 1944. The results are shown in Fig. D for the S.E. and N.W. walls. Since the thermometer used to determine the outside wall temperature was calibrated only to 120 F it was necessary to estimate the outside wall surface temperature by observation of the vibrations of the needle above 120 F. The estimated values are shown by means of a dotted line.

The surface temperatures shown in Fig. B and Fig. D were taken by means of a dermalor and graduated for studies relating to panel heating. The diagrams of Figs. A, B and D were prepared by R. G. Cox, a senior student in mechanical engineering. The hours shown in these figures are local sun time.

It may be of interest to note here that during a recent study of a building having exterior walls of limestone, painted black on a base course 4 ft high and painted white on the upper portion, the surface temperatures at 4:00 p. m. were 120 F on the surface painted black, and 106 F on the surface painted white. The temperature of the air one inch in front of the wall was 94 F.

C. M. Ashley, Syracuse, N. Y.: The method which was presented by the authors is the result of several years of trying other different methods. We have tried hydraulic methods and electrical methods of analysis, and we have also attempted to measure heat transfer in actual walls.

All those methods, to some extent, have defects. The present method may seem to be a very highly mathematical and abstract approach to the problem and yet it has the advantage that it is more general in character, and permits the use of any type of wall construction to far better advantage, and, we believe, with much greater accuracy than any of the methods which have preceded it.

We have here a tool which, with proper use, will permit us to do some very practical things. Where, 15 years ago, we took the steady flow heat transfer as a basis of our summer load calculations and then applied a factor of safety in order to arrive at the values of load, we can now analyze and take advantage of it by knowing in detail what the nature of our problem is. We can insulate to better advantage. We can adjust the capacity of the system to fit the average rather than the peak load.

One other thing, the values of the volumetric specific heat, and, to some extent also, of the absorptivity of material, are at present very poorly known and it is the hope of the Committee on Research that experimental work can be carried on to obtain more exact values for these. With these data available, we shall have a body of material from which accurate wall transmission data can be obtained.

G. L. Tuve, Cleveland, Ohio.: I am glad Mr. Ashley brought up the point that there are a great many methods for analyzing periodic heat flow. Certainly the authors are to be complimented on their success with the method they have used. Perhaps, some of you saw the item in the June issue of *Electronics*, reporting that students at M.I.T. had analyzed this problem with electronic devices, and compressed the 24-hour heat-flow cycle into 1/60 of a second, reading the results with an oscillograph. Our Society has also previously sponsored cooperative research on electrical methods and other methods of heat-flow analysis.

I want to point out just one thing, that whatever method of analysis is used, there are a number of selections or assumptions to be made before the analysis is started. One must select the volumetric specific heat of the material, the conductivity, the inside surface coefficient, and the outside surface coefficient, to say nothing of the weather curve in terms of the sol-air temperature.

Our Research Committee is aware of the complication involved in this whole problem. There are five projects dealing with periodic heat flow in this list of 28 projects to which Cyril Tasker, Director of Research, has called your attention. I think the Society can do something toward showing what are the correct selections or assumptions, so that the final results of any analysis of heat flow in walls will be more accurate than is otherwise possible.

Authors' Closure: The authors wish to thank those who prepared an oral or written discussion. Regarding the remarks made by Professor Tuve, it is true that there are a lot of variables in this problem. We have only standardized two, the outdoor film coefficient of heat transfer and the inside film coefficient of heat transfer. We have tried to effect a combination of some by the introduction of sol-air temperature, so that the effects of outdoor temperature and incident solar radiation can be properly combined.

In regard to volumetric specific heat and thermal conductivity, which are the only two thermal properties of the material influencing the unsteady heat flow, it is also true that there may be a short cut there.

The apparent density of the material governs, to a considerable degree, its thermal conductivity, k, as you all know, and it also governs, to a considerable degree, its volumetric specific heat (ρc). The specific heat of most of the common materials on a weight basis—that is expressed in Btu per pound degree Fahrenheit—does not vary widely, regardless of whether the material may be mineral, vegetable, or animal in origin. In other words, the specific heat of a cellulose product is about 0.32 Btu per pound per degree Fahrenheit; for rock and rock fibre 0.2; for glass and glass fibre 0.18. So, on a pound basis, it does not matter much what the origin of the material may be. Specific heat varies between very narrow limits. Therefore, the product of thermal conductivity and volumetric specific heat depends primarily upon the apparent density in pounds per cubic foot, and a curve of the product of k and ρc plotted vs. apparent density would come very close to giving that product for materials both known at the present time and unknown or unused at the present time.

H. L. BILSBOROUGH HEADS STOKER SALES

The A. O. Smith Corporation, Milwaukee, Wis., announces the appointment of H. L. Bilsborough, widely known in the stoker industry, as Manager of Stoker Sales.

One of the pioneers in the stoker field, Mr. Bilsborough entered this business in 1928 in the Chicago market. His first connection was with the Iron Fireman Manufacturing Co., where he advanced to district representative, covering the territory of Illinois, Indiana, and Ohio. In January of 1934 he became associated with Fairbanks, Morse & Co. as sales manager of their newly formed stoker division. He was promoted to manager of the division in July of 1938 and held that position until resigning in February of this year.

Mr. Bilsborough has always been active in the affairs sponsored by the Stoker Manufacturers Association, having served as a member of the executive board. He also has been a member of the Stoker Industry Advisory Committee to the WPB since the beginning of the war.

KNOW YOUR REPAIR MAN

In the bulletin of the St. Louis Better Business Bureau among the items listed as current rackets the following warning to investigate first is given on furnace fakers: "A bad situation exists in the furnace field. The unscrupulous repair man who offers to clean your furnace and put it in shape, often takes it apart, and then discovers broken parts. (He may actually break them himself.) He regretfully declares a new furnace would be cheaper than the large repair bill which is necessary. If you do not buy his product, he may walk out, leaving the parts strewn over the floor."

Heating engineers can do a public service by warning house owners to be on their guard against itinerant repair men and high pressure salesmen.

NEW HEADQUARTERS FOR ASA

The American Standards Association is now located in the Grand Central Office Bldg., 70 East 45th St., New York 17, which provides larger quarters than were available in the Engineering Societies Building, 29 West 39th St.

In the new location there is direct access to Grand Central Station and to local transportation facilities.

CINCINNATI MEMBER ON SMOKE ABATEMENT BOARD

Willard C. Pistler, consulting engineer, Cincinnati, a member of ASHVE since 1934, has just been appointed a member of the Smoke Abatement Advisory and Appeals Board of the City of Cincinnati for a three-year term. Announcement of the appointment was made by City Manager W. R. Kellogg and the other members of the Board are: L. A. Davidson, Raymond City Coal & Transportation Corp., North Bend, who will fill a one-year term; Dr. Glover T. Keen, two-year term; and E. S. Saurbrunn of the Babcock & Wilcox Co., four-year term.

DUBRY HONORED BY ASSOCIATES

Ernest E. Dubry of Detroit, Assistant Superintendent of Central Heating for the Detroit Edison Co., was recently tendered a testimonial luncheon by his associates to commemorate 30 years of service with the Detroit Edison Co. After serving as combustion engineer at the Conners Creek Power Plant, he joined the Navy, and after his discharge returned to the service of the Edison Co.

ROBERT B. DICKSON ELECTED DIRECTOR

Announcement has been made that Robert B. Dickson, president of Kewanee Boiler Corp., Kewanee, Ill., has been elected a director, a member of the Executive Committee, and a member of the Finance Committee of American Radiator and Standard Sanitary Corp., Pittsburgh, Pa.

Donald D. Couch has been elected vice-president and general manager of sales to succeed the late Martin J. Beirn.

ARTHUR L. HALLER DIES IN ACCIDENT

Arthur L. Haller, president of the Haller Sales Co., Inc., St. Louis, met a tragic death on July 24 in a motorboat accident. Mr. Haller was born in St. Louis on July 19, 1894, and after graduation from the public schools he entered the employ of N. O. Nelson Manufacturing Co., and for three years was a designer of heating apparatus. He then entered the contracting field in estimating and design work for the Bradley Heating Co. and then for 12 years was with the Hunt Heating Co. He organized the Haller Appliance Sales Co. to specialize in gas heating applications and represented a number of nationally known firms in the St. Louis territory.

Mr. Haller joined the Society in 1920 and was a valued member of the St. Louis Chapter.

The Officers and Council have sent his family who survive their sincerest sympathy.

1945 Nominees

Committee on Research
(Three-Year Term)

R. M. Conner, Cleveland, Ohio, director of the American Gas Association Testing Laboratories, was born on July 23, 1891 at Warrensburg, Mo. He was graduated from Oregon State College in 1915 with the degree of B.S., and later he was awarded a professional degree in mechanical engineering by the same college.

R. M. Conner
Cleveland, Ohio

Following his graduation he spent ten years with the Portland Gas and Coke Co., Portland, Ore., with the exception of a period during World War I, when he served as artillery officer overseas. He served in the transportation, distribution and utilization departments of the company, and was manager of the company's meter and repair shop for a time. He later became supervisor of the company's testing laboratory.

Mr. Conner represented the Pacific Coast Gas Association as its research engineer as a member of the technical sub-committee on the investigation of gas poisoning and asphyxiation resulting from the use of gas heating appliances in Los Angeles and the vicinity. In 1924 he was awarded the Gold Medal of the Pacific Coast Gas Association for significant service as chairman of the committee which drafted the important Gas Appliance Testing Code.

Mr. Conner has been active as a member of the Baltimore Gas Reference Committee and assisted in drawing up the Baltimore Gas Appliance Ordinance which has served as a model for similar ordinances in other cities for many years.

Mr. Conner joined the Society in 1931 and for the first four years served as chairman of the TAC on Correlating Thermal Research. Since joining the Society he has served on numerous research committees; among these are the TAC on Oil and Gas Burning Devices, Radiation—Direct and Indirect, Gas Heating Equipment, and Fuels.

He is also an active member of the *ASME, ASTM,* and is a member of Phi Kappa Phi, Tau Beta Phi, and Sigma Xi.

* * *

John A. Goff, Philadelphia, Pa., was born October 19, 1899 at Colorado City, Colo., and educated

J. A. Goff
Philadelphia, Pa.

in Colorado schools. He received his B.S., M.S. and Ph.D. from the University of Illinois, Urbana, Ill., in 1921, 1924 and 1927 respectively. While at the University he served as assistant instructor and associate professor in mechanical engineering.

He served as geophysicist for the Sun Oil Co., Dallas, Tex., and as engineer for the Linde Air Products Co., New York and Buffalo, and in 1929 joined the staff of Westinghouse Research Laboratory, East Pittsburgh, Pa., as research engineer and consultant on thermodynamics. Returning to the University of Illinois in 1930 he became associate professor of thermodynamics and professor from 1935 to 1938.

In 1938 he became Dean of the Towne Scientific School and Director of Mechanical Engineering at the University of Pennsylvania, which positions he now holds.

Dean Goff joined the Society in 1938 and served as a member of its Committee on Research for three years and as a member of the present Research Executive Committee. He also served on the TAC on Psychrometry for four years and two additional years as chairman. He has served over five years on the TAC on Instruments, and is a member of the current TAC on Sorbents.

He is the author of *Notes on Thermodynamics,* and of Chapter 1—Thermodynamics of Air and Water Mixtures in the Society's Guide, 1941. Dean Goff, who is listed in *Who's Who in Engineering,* has published numerous articles and papers in the field of thermodynamics in the Society's Transactions, as well as in *Industrial & Engineering Chemistry, American Journal of Mathematics, ASME Transactions, Refrigerating Engineering, The Technograph, Journal of Applied Mechanics,* and *Heating, Piping & Air Conditioning.*

He holds membership in the *ASME* and in the *American Mathematical Society.*

* * *

F. W. Hutchinson, Berkeley, Calif., was born on April 7, 1910, at Suijun City, Calif. He attended the J.S.U. High School at Crockett, Calif., and received his B.S. from California Institute of Technology

F. W. Hutchinson
Berkeley, Calif.

in 1931. He was graduated from the University of California in 1937 with a degree of M.S., and in 1938 was awarded his M.E. degree

from the University. While at the University he did graduate work in air conditioning.

He spent five years as construction inspector for the C. & H. S. R. Co. at Crockett, and three years as mechanical engineer in designing automatic control equipment. He has also spent some time as instructor in refrigerating and air conditioning at the University of California.

He joined the Society in 1937 as an Associate Member and became a Member in 1942. He has served as a member of the Research TAC on Radiation and Comfort for a number of years.

Professor Hutchinson, who is now associate professor of mechanical engineering at the University of California, is the author or co-author of over 35 papers, articles and reference book contributions, including publications in journals of the *American Society of Refrigerating Engineers, British Heating and Ventilating Institute, Institute of Aeronautical Sciences, American Institute of Electrical Engineers, National District Heating Association, Heating, Piping and Air Conditioning Contractors National Association, Pacific Coast Gas Association,* as well as many papers published in the Society's Journal and Transactions.

Professor Hutchinson is also a member of the *ASRE, IAS,* the Tau Beta Pi Engineering Society, the Sigma Xi, and the *National Research Society,* and is listed in *Who's Who in Engineering.*

* * *

Robert K. Thulman, Washington, D. C., was born in Arlington, N. J., on December 31, 1898. After attending schools in New York he entered Massachusetts Institute of Technology, receiving his B.S. from there in 1922.

R. K. Thulman
Washington, D. C.

His first assignment was with the Socony Burner Corp., New York, as service manager, and after a year he became research engineer for the company. In 1927 he became sales engineer for the International Boiler Works Co., East Stroudsburg, Pa., and served as engineer of the company from 1931 to 1935, at which time he joined the staff of the Resettlement Administration, Washington, D. C., as heating engineer. He specialized in the design of heating systems for suburban projects at Greenbrook, and was consultant on heating for Greenbelt and Greendale. His work as engineer also covered design of residence and school heating systems for rural projects, as well as for large scale rental projects at Buckingham, Arlington, Virginia.

In 1937 he became mechanical engineer for the Federal Housing Administration, which position he now holds.

He joined the Society in 1937, and has served as a member of the TAC on Heating Requirements of Buildings from 1940 to 1944, and as a member of the Joint Committee on Standards for Residence Heating in 1941 and 1942.

Mr. Thulman was co-author with L. E. Seeley on the paper Heating Low-Cost Homes, which was published in the ASHVE Transactions, Vol. 45, 1939, as well as several articles appearing in various technical magazines.

* * *

William E. Zieber, York, Pa., director of research of the York Corp., was born on July 12, 1888, at Reading, Pa., where he received his early education, graduating from the Reading High School in 1906. He took an I.C.S. course in mechanical engineering.

After several years' experience in shops in and around Reading, he joined the staff of York Manufacturing Co. as erecting superintendent, which covered superintending erection of refrigeration and ice manufacturing plants, re-

W. E. Zieber
York, Pa.

building of machinery and building of a 60-ton ice plant and supervising the operation of the plant. After five years he became chief engineer of the Virginia Fruit Cons. Co., Roanoke, Va., and in 1919 became chief engineer for the Peoples Ice Co., Warren, Ohio.

In 1921 Mr. Zieber returned to York, Pa., and became research engineer for the York Manufacturing Co. and served as director of research laboratory from 1921-28, when he was appointed mechanical engineer for the company. In 1935 he became assistant chief engineer and in 1940 he rose to his present position as director of research.

Mr. Zieber joined the Society in 1935 and has served on many of its committees, as a member of the TAC on Cooling Load in Summer Air Conditioning in 1942, and as its chairman for the past two years. He is among those listed in *Who's Who in Engineering.*

51st ANNUAL MEETING AT BOSTON

The invitation of the Massachusetts Chapter to have the Society's 51st Annual Meeting in Boston was accepted by the Council of the Society, it is announced by Pres. S. H. Downs.

The meeting will be held during the fourth week in January at Hotel Statler, Boston, and the Committee on Arrangements appointed by D. M. Archer, president of the Massachusetts Chapter, is now working on the program. January 22, 23 and 24 will be devoted to business and technical sessions and complete details will be found in later issues of the Journal.

SUMMARY OF LOCAL CHAPTER MEETINGS

Chapter	Meeting Date	Subject	Speakers	Other Features	Attendance	Attendance Ratio*
Delta	July 19	Progress of Chapter During Past Year	L. V. Cressy, Pres.	Installation of Officers Treasurer's Report by J. J. Friedler, Jr.	18	0.47
Washington, D. C.	May 10	Postwar Use of Electronics in Air Conditioning	J. E. Haines, Vice-Pres., Minneapolis - Honeywell Regulator Co., Minneapolis, Minn.	New officers introduced by retiring Pres. S. L. Gregg	65	0.42
Western Michigan	May 23	Election of Officers		Installation of Officers arranged by John Dykman and O. D. Marshall	29	0.88

*The attendance ratios shown represent the meeting attendance divided by the Chapter membership. These will be useful as a partial indication of interest shown by local chapter members in various types of subjects programmed by other local chapters and may be helpful in deciding on subjects for other chapter meetings.

Candidates for Membership

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the Journal of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 58 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by September 15, 1944, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
Angell, Lawrence D., Htg. & Vtg. Co-ordinator, Sperry Gyroscope Co., Inc., Brooklyn, N. Y.	A. F. Hinrichsen W. E. Heibel	R. H. Wasson F. D. Lawrence
Angelo, Herbert G., Htg. Engr., Stearns-Roger Mfg. Co., Denver, Colo.	E. J. McEahern F. L. Adams	J. J. Johnson G. D. Maves
Beline, Martin B., Engr., Washington Refrigeration Co., Washington, D. C.	P. H. Erisman, Jr. R. F. Lauer	H. Peacock H. W. Meinholtz
Bleier, Frank P., Research & Devel. Engr., Ilg Electric Ventilating Co., Chicago, Ill.	A. G. Sutcliffe J. M. Frank	B. L. Casey *U. R. Furst
Berry, Norton E., Director of Research, Servel Inc., Evansville, Ind.	W. F. Friend A. E. Vroome	M. F. Blankin A. E. Kriebel
Brown, Ward A., Asso. Engr., George Wagschal Associates, Detroit, Mich.	F. L. Pearson G. H. Giguere	R. S. M. Wilde E. F. Hyde
Bryce, John H., Supervising Engr., Kaighin & Hughes, Toledo, Ohio.	*S. D. Bryce E. W. Gray	L. E. Slawson William Goodman
Burrows, Austin J., Post Engr., Postmaster, New York, N. Y.	W. B. Leland W. W. Murphy	*Otto Epstein *C. Eddy
Campion, Charles R., Partner, Wrightson & Campion, New York, N. Y.	W. F. Wrightson C. S. Pabst	W. E. Heibel E. E. Adams
Carter, Henry G., Mfrs. Agent, Tampa, Fla.	J. C. Trefts, Jr. B. A. Thomas	O. J. Prentice F. J. Beckwith
Criqui, Albert E., Mech. Engr., Buffalo Forge Co., Buffalo, N. Y.	C. C. Cheyney W. R. Heath	R. D. Madison R. Farnham
Darby, Thomas E., General Service Mgr., The Ohio Fuel Gas Co., Columbus, Ohio.	W. M. Myler, Jr. J. D. Slemmons	A. W. Williams H. R. Allonier
Davidson, James W., Pres., Davidson Heating, Val Morin, Que., Canada.	R. A. Standing F. A. Hamlet	A. F. Lamontagne Leo Garneau
Davis, Wayne M., Field Engr., Conco Engineering Works, Muncie, Ind.	W. C. Bevington G. B. Supple	C. H. Hagedon G. O. Jackson
Englehart, Oscar D., Research Engr., Pittsburgh Plate Glass Co., Creighton, Pa.	S. F. Cox R. A. Miller	*Leighton Orr *L. T. Black
Evans, Delmer C., Chief Engr., I. Magnin & Co., Los Angeles, Calif.	J. L. Blake J. F. Park	A. J. Hess N. N. Leas
Foit, William R., Engr., Curtiss-Wright Corp., Airplane Div., Robertson, Mo.	A. L. Haller D. J. Fagin	B. L. Evans R. H. Kremer
Forsyth, Stuart L., Advisory Engr., Westinghouse Electric & Manufacturing Co., East Pittsburgh, Pa.	G. W. Penney E. H. R. Pegg	Frank Thornton, Jr. G. H. Woodard
Gelert, Carl R., Owner, C. R. Gelert Co., Los Angeles, Calif.	W. S. Kilpatrick Leo Hungerford	H. M. Hendrickson Maron Kennedy
Glanzer, Clarence J., Chief Engr., Air-Maze Corp., Northfield, Ohio.	W. B. Watterson Jules Gratiot	C. A. McKeeman D. L. Taze

*Non-Member.

GOEBUCH, WINFIELD J., Asso. Naval Archt., Norfolk Navy Yard, Portsmouth, Va.

GOTTSCHALK, KENNETH A., Indus. Sales Engr., Cleveland Electric Illuminating Co., Cleveland, Ohio.

GROSS, VICTOR L., Sales Engr., Taylor Forge & Pipe Works, New York, N. Y.

HENENFELD, HENRYK, Air Cond. Engr., A.S.E.A. Electric Ltd., Bombay, India.

HOUSEHOLDER, H. L., Mgr. & Owner, Householder Heating Corp., Buffalo, N. Y.

JANITZ, ROBERT H., Co-Partner, Fred G. & Robert H. Janitz, Indianapolis, Ind.

KEMP, W. HAMILTON, U. S. Navy, Washington, D. C.

KITTLESON, HOWARD B., Supt. of Maint., Kraft Cheese Co., Albany, Minn.

LASLEY, JAMES B., Asst. Foreman, Plant Engineer's Dept., North Carolina Shipbuilding Co., Wilmington, N. C.

LOCKHART, HAROLD A., Chief Engr., Bell & Gossett, Morton Grove, Ill. (*Advancement*)

MAC FERRIN, JOHN B., Htg. & Vtg. Engr., Pittsburgh Water Heater Sales Co., Houston, Texas.

MCFADDEN, H. J., Sales Engr., Servel Inc., Dallas, Texas.

MONTESANO, FRANK P., Mgr., The Frank A. McBride Co., Paterson, N. J. (*Advancement*)

MUSE, M. H., Mgr., Southern Welding Co., Johnson City, Tenn.

OLSON, NAT I., Owner, United Blower Co., Inc., New York, N. Y.

PIRRUNO, RICHARD W., Supvr., The Huffman-Wolfe Co., Columbus, Ohio.

POLLOCK, CLANCY W., Research Engr., Drayer & Hanson, Inc., Los Angeles, Calif.

REES, HAROLD R., Owner, Rees Plumbing & Heating Co., Jonesboro, Ark.

ROMPEL, WALTER A., Refrig. Mech., Post Engineers, Randolph Field, Texas.

RUFF, ALONZO W., Vice-Pres., York-Shipley, Inc., York, Pa.

SANGER, ERNEST, Mfrs.' Agent, Watts Regulator Co., Detroit, Mich.

SCHAAD, FRED N., Dist. Repr., Warren Webster & Co., Columbus, Ohio.

SCHMIDT, CARL W., Dist. Indus. Mgr., B. F. Sturtevant Co., Camden, N. J.

SCOTT, RICHARD P., Comm'l. Sales Mgr., Bard, Inc., Columbus, Ohio.

SIMMONS, NOAH M., Htg. & Piping Contractor, Indianapolis, Ind.

SKIDMORE, JOHN G., Constr. Engr., Almirall & Co., Inc., New York, N. Y. (*Advancement*)

SPURLOCK, B. H., Jr., Asst. Prof. of Mech. Engrg., University of Colorado, Boulder, Colo.

THACKER, JOHN E., Htg. & Vtg. Engr., Vauxhall Motors Ltd., Luton, Bedfordshire, England.

TUTTLE, HOWARD W., Design Engr., Harrop Ceramic Service, Columbus, Ohio.

WALKER, T. R., Columbus, Ohio.

WARD, BENJAMIN G., Mech. Engr., U. S. Engineers, District Office, Denver, Colo.

WATKINS, J. Z., Comm'l. Gas Sales Mgr., Rulane Gas Co., Charlotte, N. C.

WILLIAMS, JAMES K., Supt. of Constr., Robert S. Leiby Sheet Metal Co., Columbus, Ohio.

WILLIAMS, KENNETH O., Asst. Mgr., N. O. Nelson Co., St. Louis, Ill.

WINDSOR, WILLIAM G., Elec. Engr., South African Railways, Airways & Harbors Administration, Johannesburg, South Africa.

WOLFE, ALLEN H., Vice-Pres., The Huffman-Wolfe Co., Columbus, Ohio.

WOLFE, CLARENCE E., Sr., Chief Supt., Reynolds Corp., Chicago, Ill.

YOUNGBLOOD, CLETE W., Owner, Youngblood Plumbing & Heating Supply Co., Paducah, Ky.

G. E. Williams
E. B. Cary

Cyril Tasker
S. R. Guilbert

E. W. Verity
W. W. Murray, Jr.

*A. J. Dangola
*G. Berg

G. E. Adema
A. W. Wriedon

G. M. Williams
J. T. Hardin

R. H. Feltwell
M. F. Hoppe

H. C. Oberg
W. McNamara

Karl Selden, Jr.
C. Z. Adams

E. J. Gossett
C. E. Pullum

F. C. Brandt
A. B. Banowsky

J. P. Ashcraft
C. C. Young

J. Wheeler, Jr.
H. B. Matzen

Elmer Torok
*A. Roesel

V. J. Cucci
D. G. Milligan

H. Allonier
J. D. Siemmons

W. F. Walker
J. B. Griffith

W. E. Thorpe
J. J. Nolan

F. E. Giesecke
A. J. Rummel

J. J. Mirabile
E. R. Walsh, Jr.

R. L. Deppmann
W. H. Old

J. D. Siemmons
H. R. Allonier

G. P. Dickson
A. C. Bartlett

J. D. Siemmons
R. B. Breneman

W. C. Bevington
G. B. Supple

W. A. Grant
E. H. Dafter

*N. A. Parker
*W. F. Mallory

J. W. Cooling
Walter Harding

J. D. Siemmons
R. B. Breneman

W. M. Myler, Jr.
J. D. Siemmons

G. D. Maves
F. L. Adams

Karl Selden, Jr.
F. E. P. Klages

J. D. Siemmons
Howard Allonier

B. C. Simons
J. S. Malone

W. H. Carrier
W. A. Grant

H. R. Allonier
J. D. Siemmons

A. P. Brown
P. J. Marschall

C. E. Owen
W. J. MaGirl

W. C. Stubbs
C. M. Sterne

L. E. Slawson
C. M. Humphreys

L. E. Cox
S. A. Chambers

*A. F. Vitale
*J. H. Compton

F. Hockensmith
D. W. Norris, Sr.

H. V. Alexander
W. C. Bevington

A. B. DeVore
J. H. Hanlein

J. A. Craig
F. C. Winterer

R. E. Mason
F. E. P. Klages

R. E. Moore
H. B. Johns

D. M. Mills
B. P. Fisher

J. A. Ray
*Seward Abbott

D. J. Purinton
W. L. Fleisher

*A. N. Sellers
*L. E. Dequine

Ira Willner
J. J. Ferretti

R. B. Breneman
F. N. Schaad

L. Hungerford
A. Theobald

T. J. O'Brien
W. A. Danielson

*P. O'Hara
*J. M. Francis

W. E. Zieber
S. F. Nicoll

L. A. Burch
R. K. Millward

C. F. Lorimer
D. H. Wyatt

J. E. Matthews
J. A. Nicols

D. H. Wyatt
I. B. Spencer

S. E. Fenstermaker
C. H. Hagedon

W. A. Borneman
Joe Wheeler, Jr.

*C. L. Eckel
E. A. Froese

Edgar Herring
H. A. Stirzaker

D. H. Wyatt
H. R. Allonier

H. R. Allonier
A. W. Williams

D. J. McQuaid
J. H. McCabe

E. R. Harding
W. H. Sullivan

R. B. Breneman
C. F. Lorimer

George Myers
J. J. Blackmore

V. S. Day
R. V. D. Dunne

R. B. Breneman
F. N. Schaad

M. J. Bamond
O. W. Armspach

W. L. Dulle
W. K. Johnston

Candidates Elected

In the past issues of the Journal of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

MEMBERS

BARTHELMESS, RICHARD, Mech. Engr., War Department, U. S. Engineers, Jacksonville, Fla.

BEACH, ALBERT F., JR., Engr., U. S. Engineer Office, New York, N. Y.

BOYER, C. L., Consult. Engr., Dillon, Mont.

BROWNE, LELAND W., Exec. Vice-Pres., The Darby Corp., Kansas City, Mo.

BURNEY, L. BYRON, Chief Engrg. Draftsman, Norfolk Navy Yard, Portsmouth, Va.

BUSENLENER, LOUIS V., Mgr., Air Cond. Div., Higgins Industries, Inc., New Orleans, La. (*Advancement*)

CARTER, CHESTER A., Asso. Combus. Engr., Hq. 7th Service Command, R & U Branch, Omaha, Nebr.

CAVANAGH, PATRICK A., Mgr., The Herman Nelson Corp., Philadelphia, Pa.

CHESAREK, JOSEPH H., Chief, Refrig. & Vent. Sect., Office of Service Command Engineer, Hq. 9th Service Command, Ft. Douglas, Utah.

CLARK, LYNN W., Sales Engr., Hall-Neal Furnace Co., Indianapolis, Ind. (*Advancement*)

DUBE, JOHN E., Vice-Pres., Alco Valve Co., St. Louis, Mo.

ESKIN, SAMUEL G., Dir. of Research & Devel., Grayson Heat Control, Ltd., Los Angeles, Calif. (*Reinstatement and Advancement*)

EZZ-EL-DIN, KAMAL, Carrier-Egypt, S.A.E., Cairo, Egypt. (*Advancement*)

FERGUSON, C. E., Mech. Engr., Post Engineer's Office, Ft. Douglas, Utah.

FETSCHER, JOHN J., Engr., Engine Design Dept., George G. Sharp, Naval Architect, New York, N. Y.

FOX, ERNEST C., Mgr., Htg. & Cooling Dept., Hager & Cove Lbr. Co., Lansing, Mich.

FROESE, ERHARD A., Branch Mgr., York Corp., Denver, Colo.

GORDON, PETER B., Treas., Wolff & Munier, Inc., New York, N. Y. (*Advancement*)

GOULDBOURNE, THOMAS H., Sales Director, The Standard & Pochin Bros. Ltd., Leicester, England.

GRITTON, EARL V., Mech. Engr., Field Section Engineer Office, U. S. Engineers, Salt Lake City, Utah.

HALL, ELMO, Owner, Empire Gas & Equipment Co., Denver, Colo.

HERMAN, HARRY H., Owner, Harry H. Herman, Denver, Colo.

HOLFORD, DOUGLAS B., Branch Mgr., Johnson Service Co., Salt Lake City, Utah.

HOLMES, OLIVER A., JR., Mech. Engr., Hughes Heating Co. and Pritchard Bros., Memphis, Tenn.

JESPERSEN, SOREN J., Mech. Engr., Hq. 9th Service Command, Ft. Douglas, Utah.

JOHNSON, JAMES J., Htg. & Vtg. Engr., American Radiator & Standard Sanitary Corp. Denver, Colo.

KEATTS, ROLLA M., Field & Sales Engr., Russell R. Gannon Co., Cincinnati, Ohio.

KLINE, CLAYTON F., Vice-Pres., Nicholson Inc., Durham, N. C.

KNIGHTON, JOHN K., Mgr., Air Cond. Div., Servel Inc., Evansville, Ind.

KUGELER, H. COORS, Application Engr., The Stearns Roger Manufacturing Co., Denver, Colo.

LARIMER, WILLIAM M., Mgr., Htg. Dept., Crane O'Fallon Co., Denver, Colo.

LAWS, CHARLES W., Plant Supvr. of Htg., Post Engineer Office, Rapid City Army Air Base, Rapid City, S. D.

LESTER, W. L. G., Vice-Pres., Abbott Lester & Co., Inc., New York, N. Y.

LOCKE, C. F. A., Branch Mgr., Minneapolis-Honeywell Regulator Co., Indianapolis, Ind.

MATHIS, ALEXANDER, Mech. Engr., Reynolds, Smith & Hills, Jacksonville, Fla.

METCALF, RALPH H., Zone Supvr., Minneapolis-Honeywell Regulator Co., Cleveland, Ohio. (*Advancement*)

MONIES, ELLIS A., Htg. Engr., Eastern Pennsylvania Supply Co., Dallas, Pa.

MURDOCK, CHARLES E., Plant Engr., The Stokermatic Co., Salt Lake City, Utah.

NOLL, MICHAEL P., Air Cond. Sales Engr., Servel Inc., Houston, Texas.

NORMAN, EDWARD A., JR., Design Engr., Surface Combustion, Columbus, Ohio.

PETERSON, JOHN E., Tech. Adviser, Minneapolis- Honeywell Regulator Co., Chicago, Ill.

POTTER, CHRISTOPHER D., Supt. & Engr., The Smith-Gibbs Co., Providence, R. I.

POTTS, AMOS P., Chief Engr., Logan Clay Products Co., Logan, Ohio.

ROTTMAN, GEORGE B., Engr., W. H. Sullivan Co., Inc., Greensboro, N. C.

SPENCER, CHARLES H., Sales & Service Supvr., Stokermatic Co., Salt Lake City, Utah.

STILWELL, RODNEY F., Fuel Engr., Red Jacket Coal Sales Co., Columbus, Ohio.

SULLIVAN, WILLIAM H., JR., Vice-Pres., W. H. Sullivan Co., Inc., Greensboro, N. C.

TASH, THOMAS, The Kemper Lane Hotel, Cincinnati, Ohio. (*Reinstatement*)

THEISS, ERNEST S., Instructor in Mech. Engrg., Duke University, Durham, N. C. (*Advancement*)

TRAUTMAN, FRED L., Owner, Trautman Engineering Co., Denver, Colo.

VANDERHOOF, AUSTIN L., Dist. Repr., Warren Webster & Co., Cleveland, Ohio. (*Advancement*)

WALSH, JOHN W., Pres. & Treas., John W. Walsh, Inc., Worcester, Mass.

WALTERS, THOMAS A., Director of Research, American Blower Corp., Detroit, Mich.

WARD, CYRIL B., Supt., Gas Htg. Dept., Public Service Company of Colorado, Denver, Colo.

WARD, JOSEPH A., Pres., Abbott Lester & Co., Inc., New York, N. Y.

WATERMAN, DOUGLAS R., Sales Engr., Hendrie & Bolthoff Manufacturing & Supply Co., Denver, Colo.

WHYTE, FREDERIC J., Tech. Advisor, Norris Warming Co., Ltd., London, England.

WITHERIDGE, WILLIAM N., Chief Engr., Bureau of Industrial Hygiene, Detroit Department of Health, Detroit, Mich.

WOEHLKE, HERMAN J., Estimator & Asst. Mgr., Bell Plumbing & Heating Co., Denver, Colo.

YEAGER, GEORGE F., 1st Lt. T. C., U. S. Army, c/o Postmaster, New York, N. Y. (*Reinstatement*)

ASSOCIATES

ALEXANDER, JOHN L., Owner, C. H. Alexander & Son Co., Grand Rapids, Mich.

ANDERSON, EDWIN C., Owner, Edwin C. Anderson Co., Denver, Colo.

ANDERSON, ROBERT B., Owner & Mgr., Robert B. Anderson, Denver, Colo.

ASHTON, JED L., Owner, Ashton Heating & Air Conditioning, Salt Lake City, Utah.

BAKKO, REUBEN, U. S. Navy, % Postmaster, San Francisco, Calif.

BALCH, ROBERT M., JR., Sales Mgr., Apartment Controls Div., Minneapolis-Honeywell Regulator Co., Minneapolis, Minn.

BARNES, JOHN C., Branch Mgr., National Radiator Co., Washington, D. C.

BATEMAN, WILLIAM H., Partner, Keystone Sales Co., Pittsburgh, Pa.

BELL, JOHN A., Royal Canadian Air Force, No. 2 Training Command, Winnipeg, Man., Canada.

BERNZEN, JOHN, Treas., City Plumbing & Heating Co., Boulder, Colo.

BRAGG, ROBERT E., Mfr's. Repr., Indianapolis, Ind.

BREN, LEO W., Asst. Mgr., Grudem Brothers Co., St. Paul, Minn.

BROWN, EDWARD A., Mfrs.' Repr., Omaha, Nebr.

BROWN, NORMAN, Owner, Norman Brown Co., San Francisco, Calif.

BUCKINGHAM, ROBERT B., Partner, Associated Southern Industries, Memphis, Tenn.

CALCATERRA, LOUIS A., Mgr., Stoker & Htg. Div., Bennett Fuel Co., Grand Rapids, Mich.

CODERE, JEAN-MOISE, Director, Codere Ltd., Sherbrooke, Que., Canada.

COOK, CLIFFORD B., Partner, F. G. Cook & Sons, Kenmore, N. Y.

COOK, VERNON D., Partner, F. G. Cook & Sons, Kenmore, N. Y.

COOKE, WILLIAM L., Mgng. Director, W. L. Cooke Ltd., Auckland, S. E., New Zealand.

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